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FINAL REPORT

ORBIT-TO-ORBIT SHUTTLE ENGINE DESIGN STUDY

Contract F04611-71-C-0040

BOOK 2

W. P. Luscher, et. al.
Aerojet Liquid Rocket Company
Sacramento, California

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May 1972

Air Force Rocket Propulsion Laboratory
Edwards Air Force Base, California

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FINAL REPORT

ORBIT-TO-ORBIT
SHUTTLE ENGINE DESIGN STUDY

W. P. Luscher, et. al.
Aerojet Liquid Rocket Company
Sacramento, California

May 1972

Headquarters, Air Force Flight Test Center
Rocket Propulsion Laboratory
Edwards Air Force Base, California

FOREWORD

This report presents the work accomplished on Contract F04611-71-C004, the Orbit-to-Orbit Shuttle Engine Design Study (OOS) over the period from 1 March 71 to 1 December 1971. The program was admistered by the Procurement Division of the Directorate of Material, Edwards Air Force Base, Edwards, California. The technical project manager at the Rocket Propulsion Laboratory, Edwards, California was Mr. L. Tepe. Mr. Werner P. Luscher directed the study effort for Aerojet Liquid Rocket Company.

This report is contained in 4 books described as follows:

Book 1: Parametric Cycle Study

Book 2: 25K lb Engine Design

Book 3: 25K lb Engine Maintenance, Development Plans, Cost Estimates and 10K lb Engine Design

Book 4: Appendices

This technical report has been reviewed and is approved.

L. E. Tepe
Project Manager

ABSTRACT

This report presents the analytical design of propulsion systems utilizing LOX/Hydrogen propellants to be used as the propulsion for the Orbit to Orbit Space Vehicle of 65,000 lb lift-off weight.

The report contains the evaluation of various engine cycles in the thrust range of 8,000 lb to 50,000 lb thrust for performance, weight and envelope culminating in the cycle selection and detail design of a 25,000 lb and 10,000 lb thrust engine. The engine concepts are described in sufficient detail to obtain reliable engine weight, performance, envelope information and methods of engine control. The impact of various engine design requirements were evaluated. The engines are designed to be reusable and capable of starting in the idle mode operation.

The technology requirements for meeting the engine design and operating requirements are identified.

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NOMENCLATURE

UNITS:

°F	Degrees Farenheit
ft	Feet
GPM	Gallons per Minute
hr	Hours
HP	Horsepower
in.	Inches
lb	Pounds
mm	Millimeters
psi	Pounds per Square Inch
rpm	Revolutions per Minute
°R	Degrees Rankine
sec	Seconds

SYMBOLS:

A/R	Area Ratio
CR	Contraction Ratio
DN	Bearing (Bore Diameter mm x rpm)
ϵ	Area Ratio
F	Thrust
I_s	Specific Impulse
L'	Chamber Length
MR	Mixture Ratio
M	Mach Number
MRD	Mixture Ratio Distribution
Ns	Specific Speed
N_f	Number of Cycles
P_c	Chamber Pressure
PVC	Pressure Volume Compensated
SF	Safety Factor
S	Suction Specific Speed
T_{T_i}	Turbine Inlet Temp
T_{FD}	Fuel Pump Discharge Temp
[illegible]	Chamber Hot Wall Temp
ΔT_W	Chamber Wall Temp Gradient
T_B	Bulk Temperature
V	Velocity
$\dot{W}$	Weight Flow Rate
W_{BO}	Burnout Weight
[illegible]	Efficiency

GLOSSARY:

ALRC	Aerojet Liquid Rocket Company
AGCarb	Carbon Cloth Material
CJKT	Preburner & Turbine By-pass
ERE	Energy Release Efficiency
FTPA	Fuel Turbopump Assembly
FDV	Fuel Discharge Valve
FRHG	Fuel Rich Hot Gas
FPBYV	Fuel Preburner & Turbine By-Pass Valve
FFC	Final Flight Configuration
Hz	Frequency
LRU	Line Replaceable Unit
N_{TF}	Fuel Turbine rpm
N_{TO}	Oxidizer Turbine rpm
N_{TFE}	Fuel Low Speed Inducer rpm
N_{TOE}	Ox Low Speed Inducer rpm
NPSH	Net Position Suction Head
NPSP	Net Positive Suction Pressure
OTPA	Oxidizer Turbopump Assembly
OPBV	Oxidizer Preburner Valve
OSCV	Oxidizer Mixture Ratio Control Valve
PL	Payload
PFC	Preliminary Flight Configuration
TCA	Thrust Chamber Assembly
TGJD	Temp at FRHG Injector Inlet
UTMO	Max Oxidizer Turbine Tip Speed
UTMF	Max Fuel Turbine Tip Speed

B. 25K THRUST ENGINE DESIGN

This section describes the 25K thrust engine analysis performed under the Task I of the Work Statement and includes the general engine cycle description, selected engine configuration, and detailed component design. The engine characteristics and interface requirements are defined for the selected engine configurations and control methods.

1. Engine System Design Description

It was concluded from the parametric analysis of six different engine cycles, that the stage combustion engine cycle has a decisive payload advantage at the 25K thrust level within the selected envelope and engine life constraints. Consequently it was selected as the engine cycle for the 25K engine design.

a. Nominal Engine Design Point Selection

The engine design requirements presented in the following table form the basis of the 25K engine design point selection. The parametric analysis indicated that for this thrust level a chamber pressure of 1800 psia can simultaneously meet the feed system power balance and thrust chamber low cycle fatigue life requirements. The results indicated that for this chamber pressure, the retractable nozzle concept yielded insignificant payload advantages and a fixed nozzle was groundruled. The elimination of the variable nozzle concept simplifies the engine development cost and maintenance and will increase engine reliability.

The selected injector concept permits a very fine injector pattern and analysis indicates that the thrust chamber requires a chamber length of L' = 6 in. from the injector face to the chamber throat. With this basic design groundrule and the given engine envelope length of 82 in., engine overall length, a nozzle expansion area ratio of ε = 290:1 is feasible.

The basic engine design point is summarized as follows:

SELECTED DESIGN POINT FOR 25K THRUST ENGINE

Mixture Ratio O/F	MR = 6.0:1
Chamber Pressure	P_c = 1800 psia
Nozzle Area Ratio	ε = 290:1
Throat Radius	R_t = 1.51 in.
Thrust/Chamber Pressure	F/P_c = 13.88
Nozzle Concept	Fixed Minimum Length Rao
Chamber Length	L' = 6 in.
Engine Overall Length	L = 82 in.
Maximum Engine Diameter	D_{max} = 51.42 in.
Nominal Engine Specific Impulse	I_s = 465.7 sec
Engine Spec. Imp. Tolerance	I_s = 465.7 to 469.7 sec

These design data were used to verify engine design compliance with the stated engine requirements.

b. Nominal Engine Cycle Description

The engine cycle for the baseline engine is shown in Figure 124 describing the engine flow, temperature and pressure schedules. A detailed description of the nominal turbopump operating conditions is shown in Section B,2,g and are based in the engine power balance analysis.

Thrust chamber cooling characteristics are based on a detailed heat transfer analysis and its results are summarized:

THRUST CHAMBER NOMINAL COOLING CHARACTERISTICS

Cooling Method	All regen, no barrier cooling
Area Ratio Regenerative Cooled	ϵ_o = 290
Coolant Pressure Drop	ΔP_c = 765 psia
Coolant Temperature Rise	ΔT_c = 295°F
Coolant Throat Mach No.	M_t = 0.46
Coolant Inlet Pressure	P_{in} = 4250 psia
Throat Wall Temperature	T_{wg} = 905°F
Throat Wall Temperature Gradient	ΔT_w = 865°F

In the fuel circuit, the feed system features a three stage fuel pump and a low speed inducer driven by a full flow hydraulic turbine. The fuel main pump is driven by a two stage turbine. The oxidizer circuit feed system also has a similar low speed inducer and drive. The main oxygen pump is a 1-1/2 stage design where the first stage feeds all the oxidizer to the pressure level required for the thrust chamber and the half stage feeds a fraction of the oxidizer required for the preburner.

The fuel rich preburner delivers fuel rich combustion products at 1860°R to the LO_2 and fuel turbine which are in a parallel flow system. The preburner and main injector circuit pressure drops were established for gaseous propellant injectors. The preburner oxidizer is vaporized in a vaporizer forming an integral part of the preburner chamber. The main injector oxidizer is vaporized in injector vanes by the turbine exhaust gases.

The engine control utilizes a three valve control system. At full thrust, engine mixture ratio is controlled by the chamber liquid oxygen inlet valve No. 1 and the thrust is controlled by the preburner liquid oxidizer No. 2 controlling preburner mixture ratio. In the throttled operating mode, the preburner and turbine bypass valve No. 3 is modulated from 80% to minimum thrust to control the preburner temperature.

The coolant jacket bypass valve No. 4 is only used to institute pump chilldown and is normally closed during engine operation except a very small flow to cool this valve.

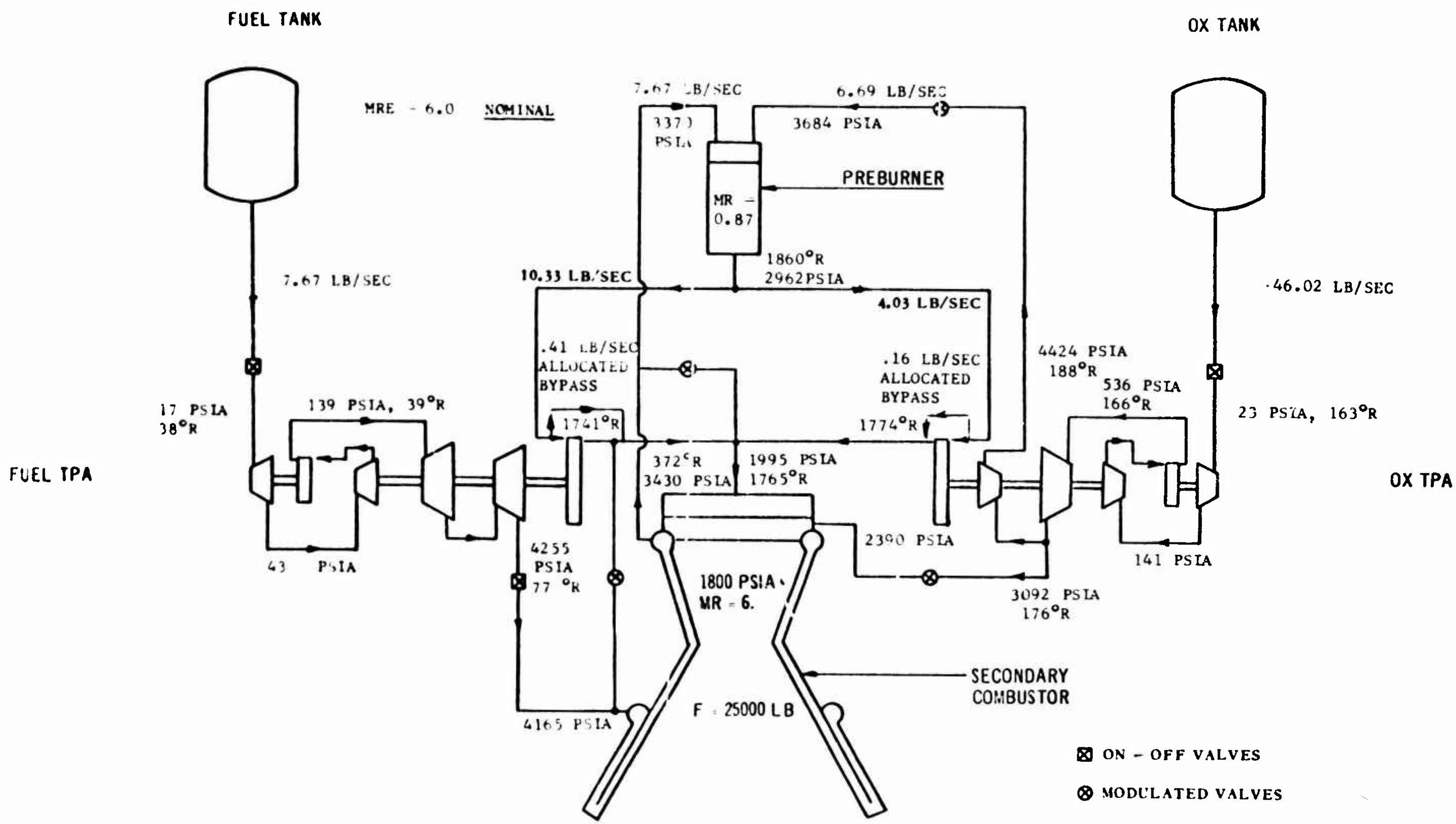


Figure 124. 25K Baseline Engine Operating Parameters

In the basic cycle shown in Figure 124, the internal leakage losses of the turbopumps are indicated by turbine bypass flows for both propellant circuits. These leakage losses are quite considerable in small pumps and effect the power balance capability considerably.

In the following, feed system nominal operating conditions are summarized:

25K THRUST FEED SYSTEM NOMINAL OPERATING CONDITIONS

	Fuel Circuit	LO_2 Circuit
Flow gpm	791.80	290.31
Inducer Pressure Rise, psia	26.40	118.3
Inducer, rpm	22,857	14,285
Inducer Specific Speed, Ns	4,000	4,000
Inducer Suction Specific Speed	12,610	25,742
Inducer Efficiency, %	71.3	66.3
Inducer HP	17.1	30.2
Mainstage Pressure Rise, psia	4,144	2,557
Mainstage, rpm	80,000	50,000
Mainstage Specific Speed, Ns	719.8	1,396
Suction Specific Speed, S	4,283	4,596
Pump Efficiency	61.5	63.80
Pump Horsepower	3,108	678
Number of Pump Stages	3	1-1/2
Turbine Inlet Temperature °R	1,860	1,860
U/C	0.426	0.255
Turbine Admission	100	31.2
Turbine Efficiency	65.5	47.40
Number of Turbine Stages	2	1

The detailed heat transfer analysis of the thrust chamber emphasized a significant fact that the coolant pressure drops requirements are also sensitive to the coolant inlet pressure. The higher the inlet pressure, the lower the coolant pressure drop requirement. This relationship is shown in Figure 125 and is considered in the engine power balance analysis.

c. Engine Control Selection

The prime selection of engine thrust and mixture ratio control valves is a preburner oxidizer valve and a main thrust chamber oxidizer valve. Figure 126 is a schematic of other valve locations considered. Additional investigation indicates that the original selection is satisfactory for an engine having a 5:1 throttling range; however, investigation of the effects of designing for a 10:1 throttling range indicate that a preburner fuel bypass valve would be a desirable addition to the oxidizer control valves. (This valve is shown as valve 9 on the schematic, Figure 126.)

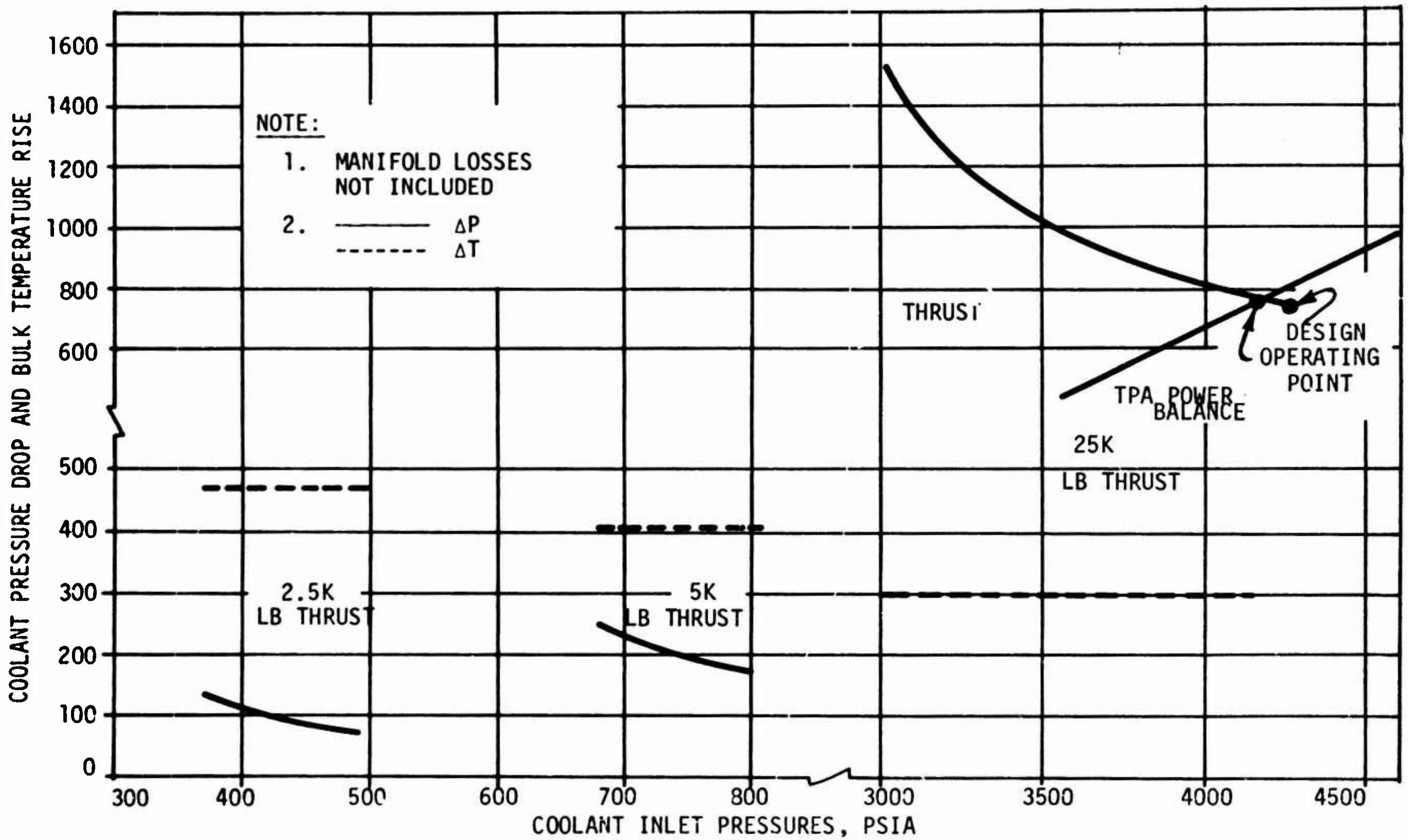


Figure 125. LH_2 Coolant Pressure Drop and Bulk Temperature Rise for Various Inlet Pressures

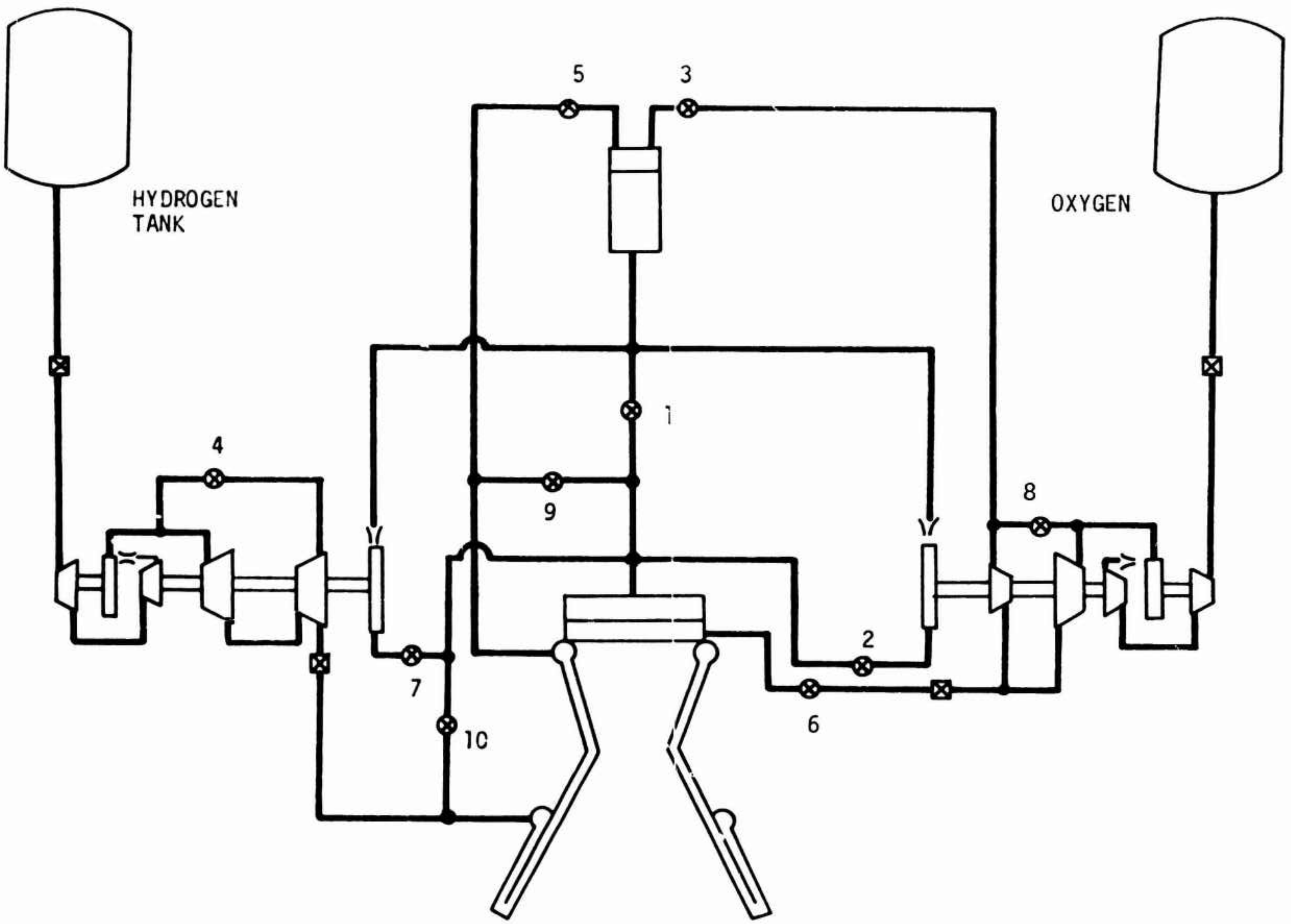


Figure 126. System Schematic Showing Candidate Control Valve Locations (Sheet 1 of 2)

THRUST CONTROL VALVES

THRUST CONTROL VALVES

Location	Nomenclature	$\frac{\Delta F/F}{\Delta X/X}$ %/%	$\frac{\Delta MR/MR}{\Delta F/F}$ %/%	$\frac{\Delta T_G}{100\ F/F}$ °R/%	Fluid Temp °R
1	Turbine Hot Gas Bypass	-1.90	-.175	-2.76	1860
2	Oxid Turbine Exhaust	-0.980	+.662	+13.5	1743
3*	Oxid Preburner Valve	-0.445	+.272	+13.7	239
4	Fuel Pump Recirculation	-0.250	-6.30	+116	110
5	Fuel Preburner Inlet	+0.206	+2.36	+49.5	428
6	Oxid Thrust Chamber Inlet	+0.184	-0.230	+17.2	204
7	Fuel Turbine Exhaust	+0.148	+6.34	+109	740
8	Oxid Pump Recirculation	-0.0825	+0.50	+7.45	204
9	Preburner Fuel Bypass	-0.675	+0.220	-17.1	446
10	Cooling Jacket Bypass	-0.550	+0.313	-19.5	120

Selection Criteria:

Thrust Control:

$\frac{\Delta F/F}{\Delta X/X}$ = Large

$\frac{\Delta MR/MR}{\Delta F/F}$ = Small

MIXTURE RATIO CONTROL VALVES

Location	Nomenclature	$\frac{\Delta MR/MR}{\Delta X/X}$ %/%	$\frac{\Delta F/F}{\Delta MR/MR}$ %/%	$\frac{\Delta T_G}{100\ F/F}$ °R/%	Fluid Temp °R
1	Turbine Hot Gas Bypass	+0.331	-5.72	15.8	1860
2	Oxid Turbine Exhaust	-0.650	+1.51	+20.3	1743
3	Oxid Preburner Inlet	-0.121	3.68	+50.4	239
4	Fuel Pump Recirculation	+1.57	-.159	+18.4	110
5	Fuel Preburner Inlet	+0.471	+.424	+21.0	428
6*	Oxid Thrust Chamber Inlet	-0.042	-4.35	-75.0	204
7	Fuel Turbine Exhaust	+0.940	+.158	+17.2	1740
8	Oxid Pump Recirculation	-0.041	+2.00	+14.9	204
9	Preburner Fuel Bypass	-0.149	+4.54	-78.3	446
10	Cooling Jacket Bypass	-0.172	+3.20	-62.5	120

Mixture Ratio Control:

$\frac{\Delta MR/MR}{\Delta X/X}$ = Large

$\frac{\Delta F/F}{\Delta MR/MR}$ = Small

Figure 126. System Schematic Showing Candidate Control Valve Locations (Sheet 2 of 2)

The main function of the preburner fuel bypass valve is to permit control of preburner mixture ratio independently of turbine power. Thus, preburner oxidizer flow rates can be higher, giving better injector stiffness and simplifying injector design. This valve is also very useful in controlling engine turbopump acceleration during the start transient. As a result, it has been added to the recommended control valve configuration for the baseline engine. It would be closed at 25K but could be opened so as to maintain preburner mixture ratios between 0.6 and 0.8 when throttling below about 85% thrust.

d. Baseline Engine Configuration Selection

(1) Baseline Description

The OOS baseline engine configuration is defined in Engine Assembly Drawing 1161635 presented as Figure 127 in this report. This design reflects the culmination of an effort involving a series of layouts, wherein an optimum configuration evolved for the 25,000 lb thrust staged combustion cycle engine. Design criteria, used as a basis for the engine assembly, are shown in Table II of the Assembly Drawing. The baseline power balance depicting the engine system steady state flow, pressures, and temperatures is shown in Figure 128.

The OOS baseline engine assembly is composed of the following major assemblies which, when assembled, produce a total engine dry weight of 459.3 lb:

Major Assembly	Weight, lb
Thrust Chamber Assembly (TCA)	178.0
Consisting of:	
Injector Assembly	
Thrust Chamber/Nozzle Assembly	
TCA Igniter Assembly	
Preburner Assembly	24.7
Hot Gas Manifold	57.7
Turbopump Assemblies	67.5
Valves	43.1
Propellant Lines	20.6
Gimbal System (Gimbal Actuator assumed separate from engine system weight) consisting of:	9.9
Gimbal Assembly	
Gimbal Actuator Support Structure	
Engine Control System and Structure (Controller assumed separate from engine system weight) consisting of:	45.5
Harnesses	
Sensors	
Support Structure	
Engine Attach Hardware and Seals	12.0
Engine Assembly Total Dry Weight	459.3

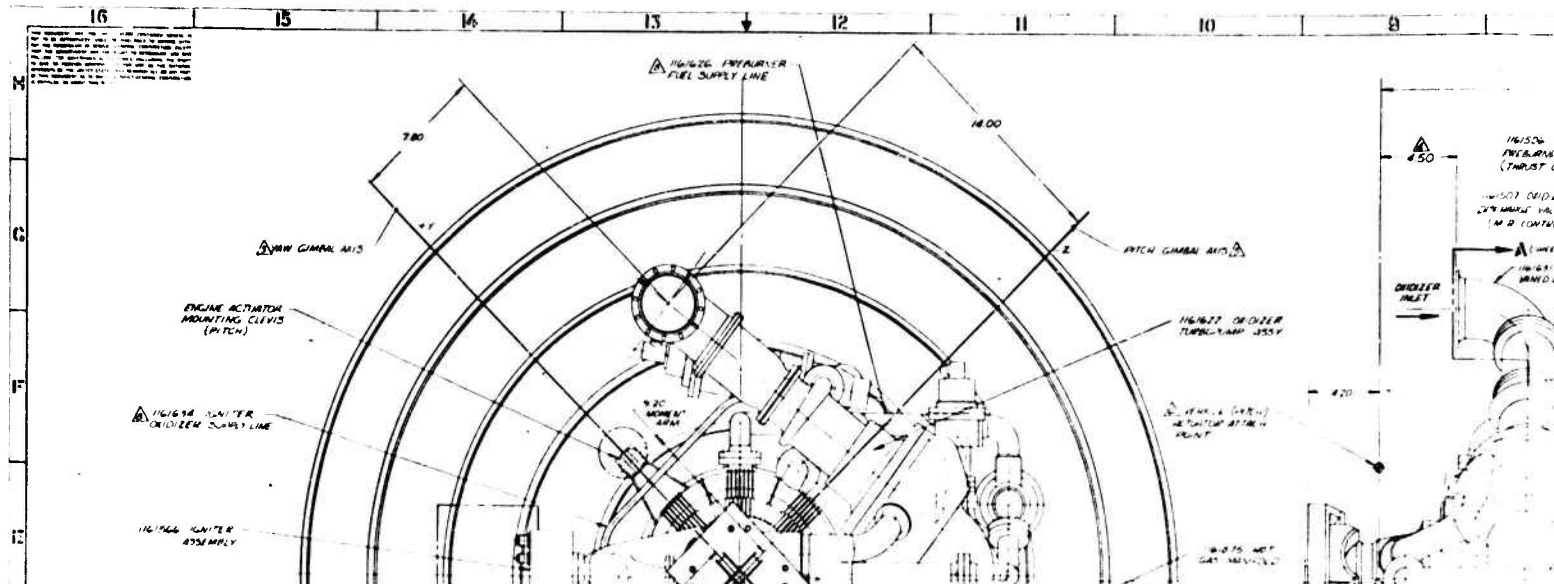
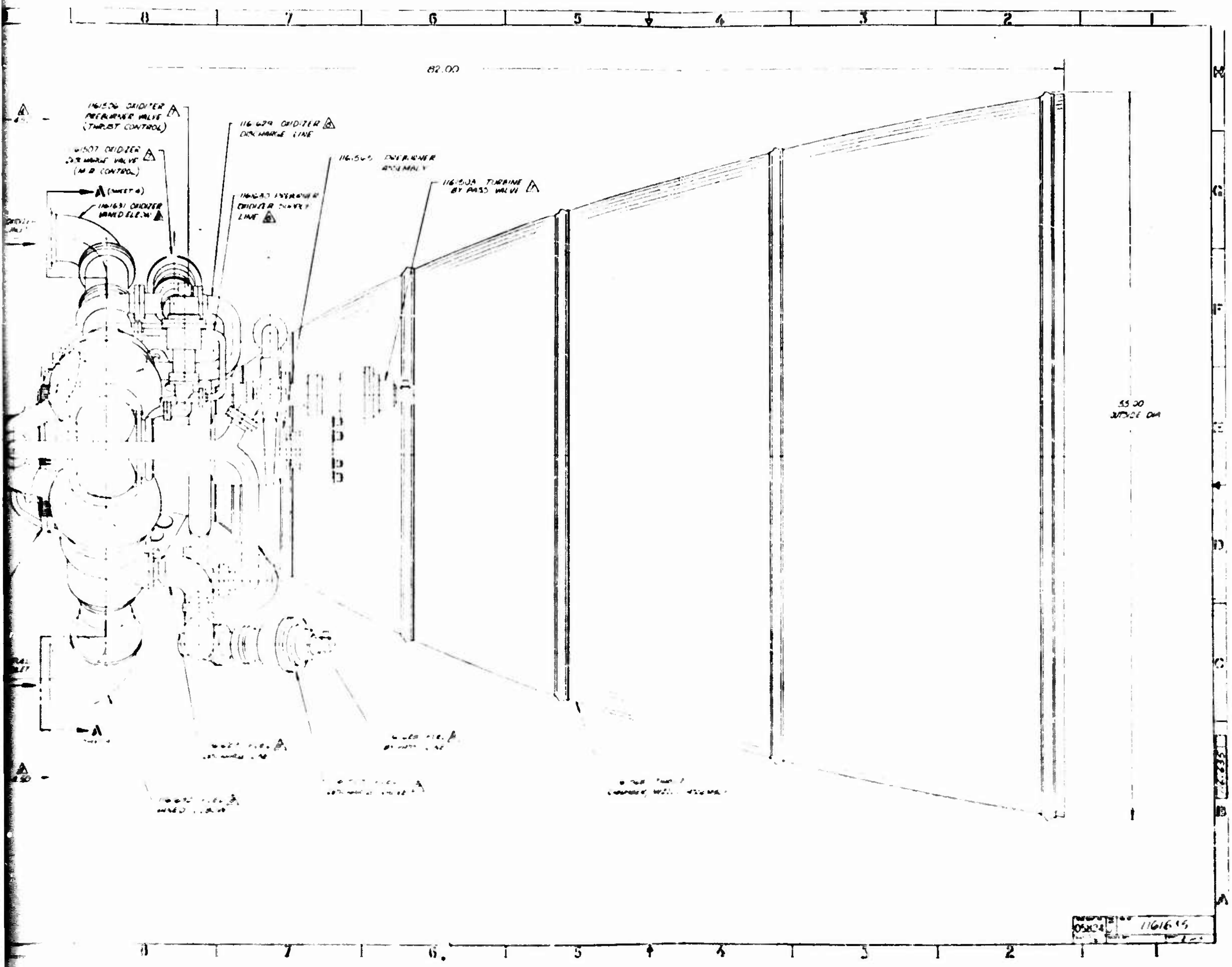


Figure 127. Engine Assemb



ine Assembly (Sheet 1 of 5)

2

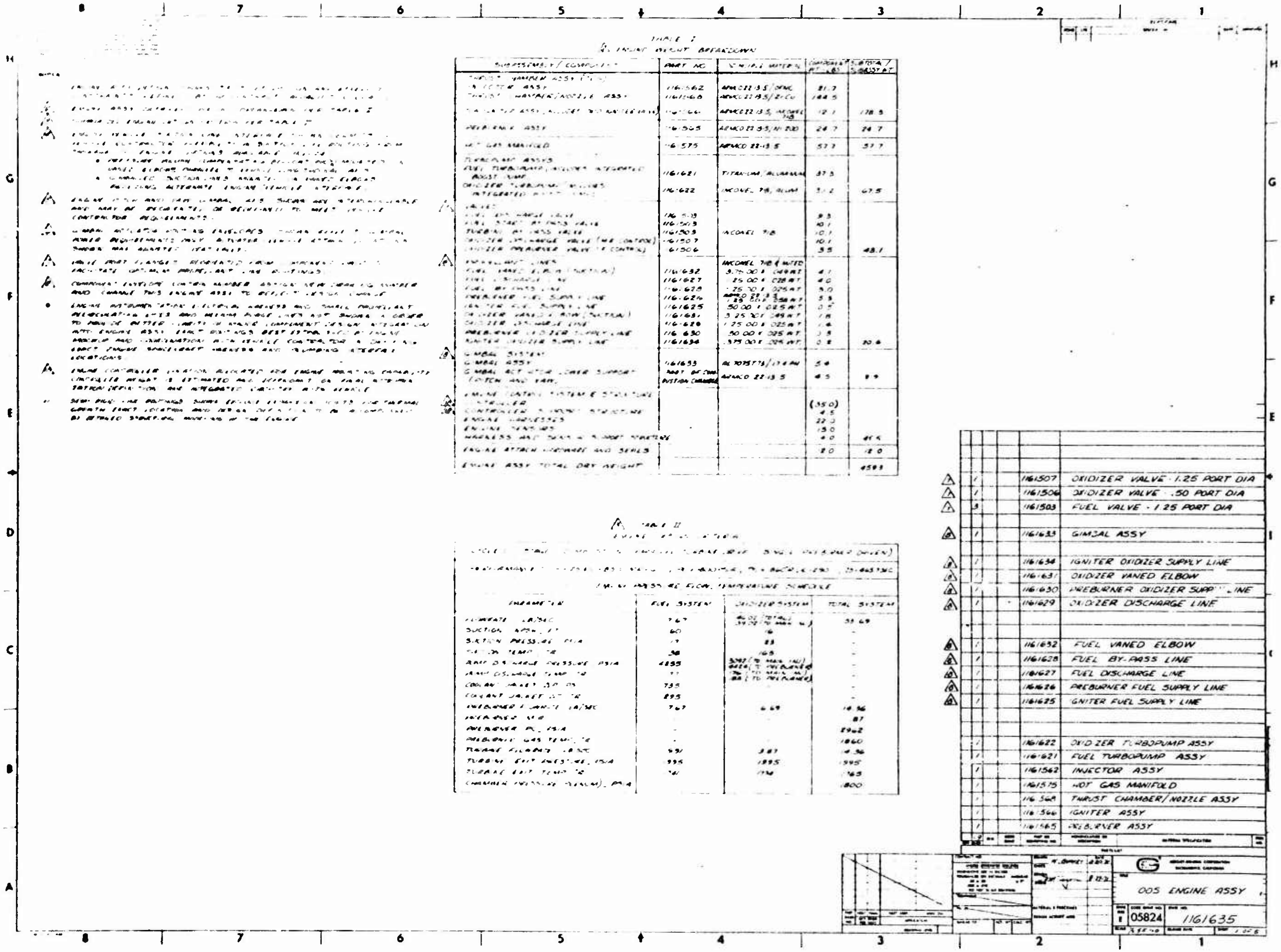


Figure 127. Engine Assembly (Sheet 2 of 5)

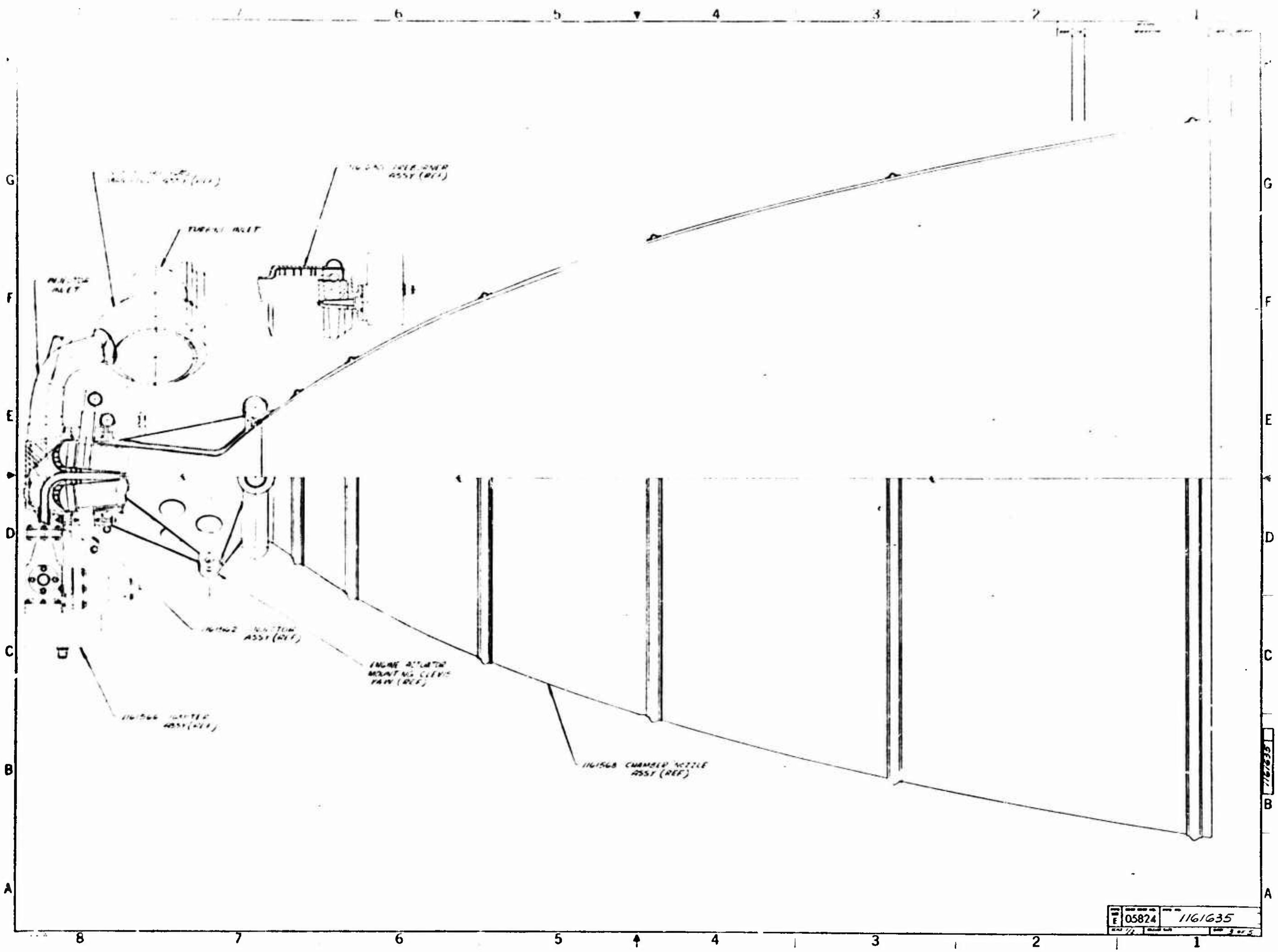


Figure 127. Engine Assembly (Sheet 3 of 5)

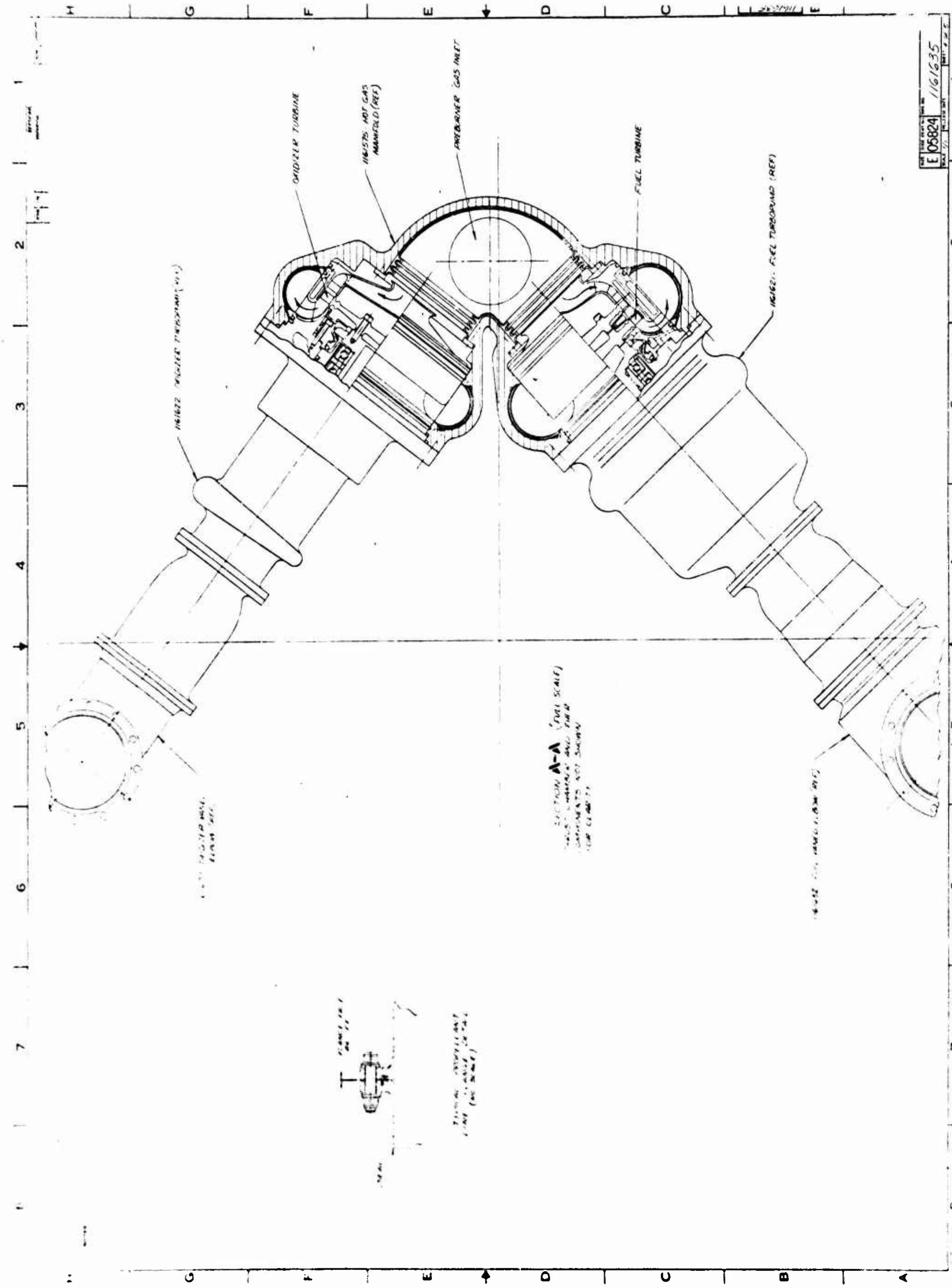


Figure 127. Engine Assembly (Sheet 4 of 5)

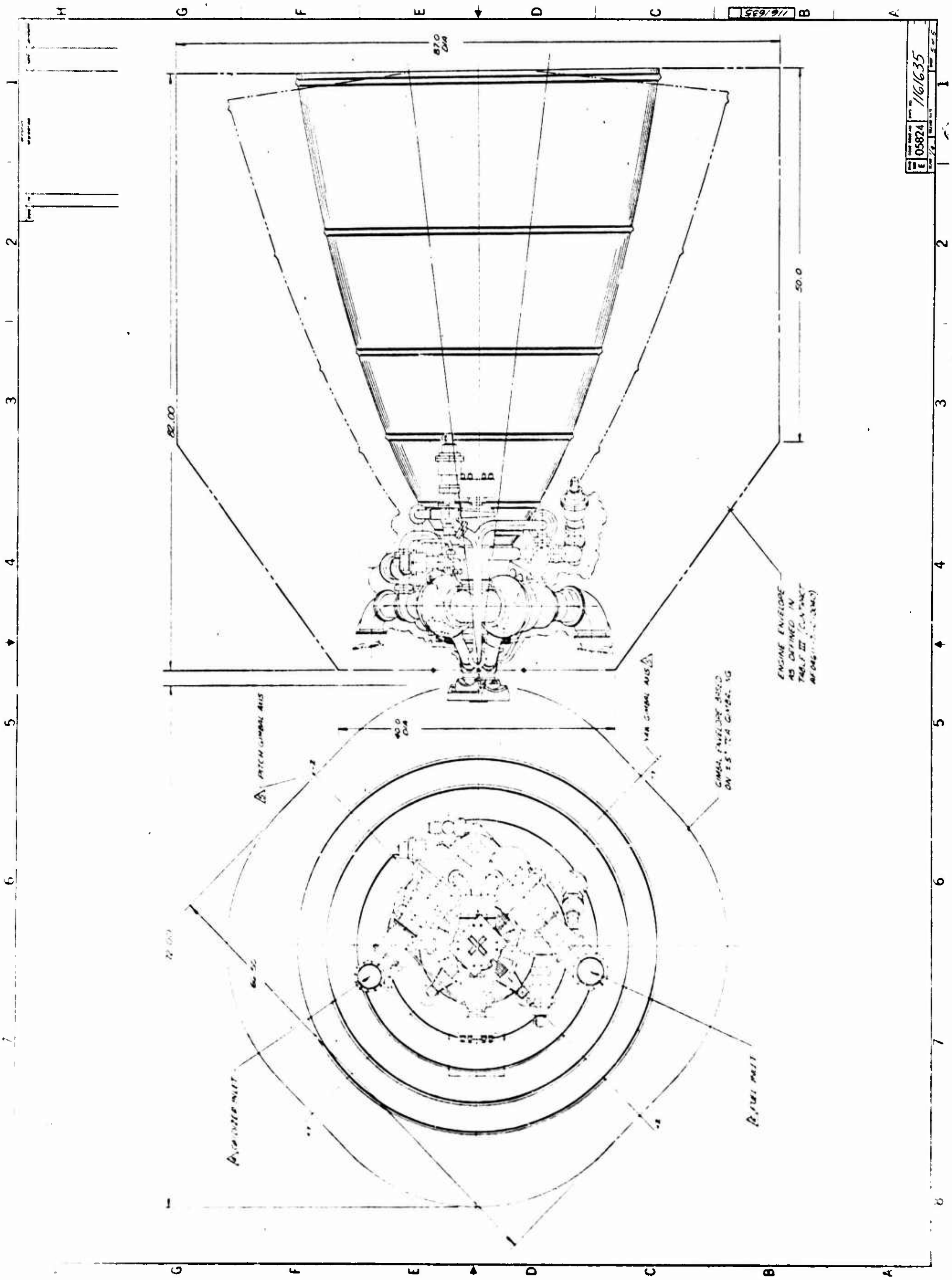


Figure 127. Engine Assembly (Sheet 5 of 5)

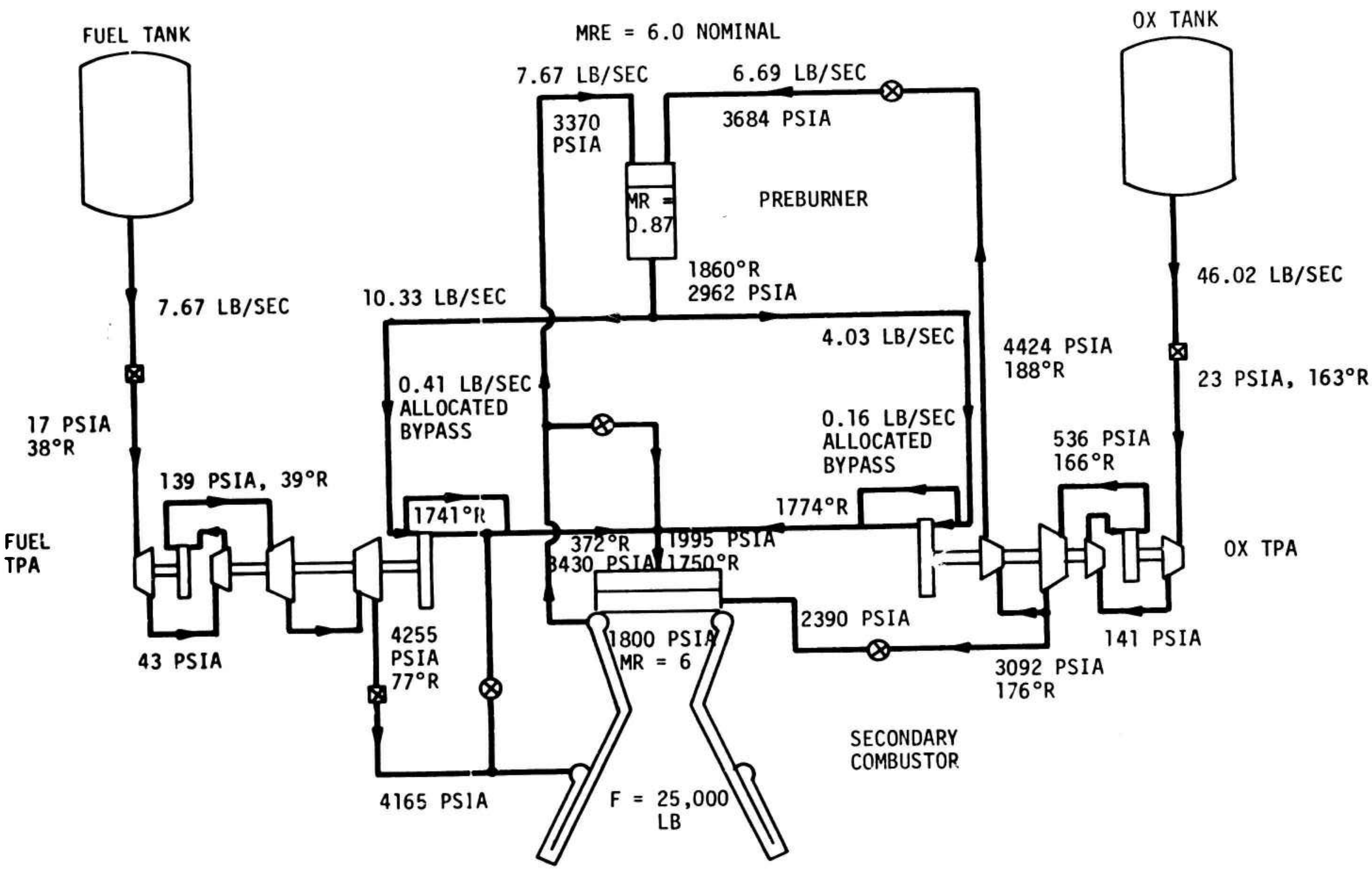


Figure 128. 25K Engine Operating Parameters

A detailed component weight breakdown including principle materials may be found on Sheet 2 of the Engine Assembly in Figure 127.

As shown in Sheet 1 of Figure 127, the hot gas manifold is the engine structural component. The high pressure turbopump assemblies (HPTPA's), including integrated in-line low pressure turbopump assemblies (LPTPA's), are side mounted laterally to an integrated turbine housing, as is the preburner. The gimbal block assembly is affixed to the upper side of the conical centerbody and the injector - thrust chamber - nozzle assembly to the lower side. The main combustion chamber igniter is also mounted to the conical centerbody of the manifold.

Propellant lines interface with the vehicle at a station plane 4.50 in. aft of the gimbal point. Suction lines, as shown, are vaned elbows which attach to the low pressure turbopump (LPTPA) inlets. Hydrogen lines and the turbopump, upstream of the fuel discharge valve, are insulated. The discharge line, downstream of the discharge valve to the chamber coolant jacket inlet, is also insulated for engine idle mode operation to minimize heat input into the low flow hydrogen.

Engine gimbaling is provided for a design pitch and yaw gimbal angle of $\pm 5°$ (including an additional 1° for snubbing, thrust alignment, and stroke tolerance). Engine gimbal compliance with the vehicle envelope is demonstrated on Sheet 5 of Figure 127. Cylindrical envelopes accommodate the vehicle-supplied gimbal actuators which are attached to the thrust chamber 9.2 in. from the engine centerline.

Engine instrumentation, electrical harnesses, small propellant recirculating lines, and helium purge lines are omitted for clarity. Packaging effort of the engine mounted controller was limited to the definition of the mounting location and estimated envelope because final controller design/size is dependent on establishment of final instrumentation and integrated circuitry with the vehicle.

(2) Trade Studies and Options

The design goals of the Task I engine configuration study were:

Engine/Vehicle Envelope Compliance.
Minimum Weight.
Minimum Engine Length.
Forward Center-of-Gravity.
Maintainability.

The results of the trade studies produced design features which permit engine design flexibility particularly in the area of engine/vehicle interface definition. These features are discussed in the following paragraphs.

(a) Powerhead Selection

Side-mounted turbopumps were selected to minimize engine length and the distance from the engine interface to the throat plane. This feature is particularly critical to a fixed nozzle engine design. Lateral, rather than vertical, turbopump mounting was selected to minimize powerhead length, move center of gravity forward, and to provide flexibility in suction line routings, vehicle interface options, and maintenance.

Selection of a single preburner, rather than duel preburners, to drive the turbines in parallel, allowed selection of an integrated turbine manifold over a separate manifold for each turbopump. Each turbopump is flange mounted to the integrated turbine manifold resulting in a lateral "V" configuration oriented about the engine/combustion chamber. The single preburner, also mounted to the manifold, provides fuel rich combustion gases at 1860°R which are directed into a common plenum chamber, ducted through the turbine, and then directed to the main injector to complete staged combustion. Section A-A on Sheet 4 of Figure 127 depicts the turbopump/turbine manifold interface and the turbine gas flow paths. Selection of the integrated turbine manifold was based primarily on:

Elimination of hot gas lines between the single preburner and separate turbine manifolds.

Predicted weight savings utilizing an integrated manifold over two separate manifolds.

Side mounted lateral "V" turbopump mounting on one side of engine permits flexibility in gimbal actuator location and suction line routing.

(b) Engine-Vehicle Interface Definition

1 Propellant Lines

Table XX presents the matrix utilized to select baseline suction line routings and related engine-vehicle interfaces. The matrix considered both engine mounted and vehicle mounted boost pumps. The selection of integrated in-line units influenced the decision on engine/vehicle suction line interface definition. As shown in Table XX, the optimum suction line routing is Option 2 which utilizes pressure volume compensating bellows (PVC's) mounted to vaned elbows which are in turn mounted to the boost pump inlets. This configuration is minimum in weight and has the least suction line pressure drop. However, to provide flexibility in suction line routing for vehicle contractors, a compromise was selected with the vaned elbow inlets at a station plane 4.50 in. aft of the gimbal point, as shown in Table XXI. Under these conditions, options are provided for direct

Table XX. OOS Suction Line Configuration Matrix (Sheet 1 of 2)

TASK I
OOS SUCTION LINE CONFIGURATION MATRIX
GIMBALLING BOOST PUMPS

CONSIDERED CONFIGURATIONS

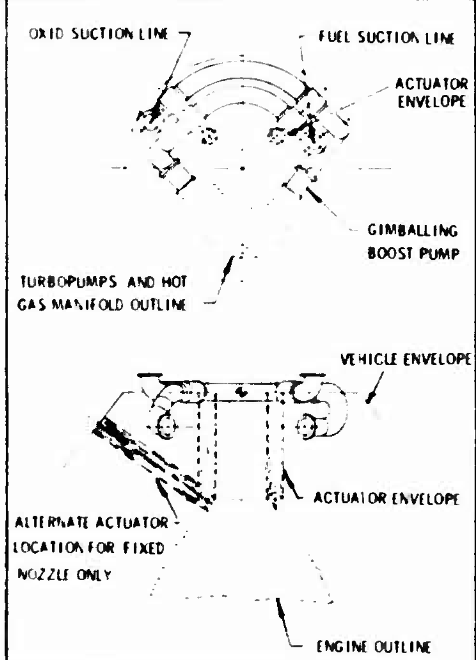
SPECIFICATIONS:

GIMBAL ANGLE ± 5

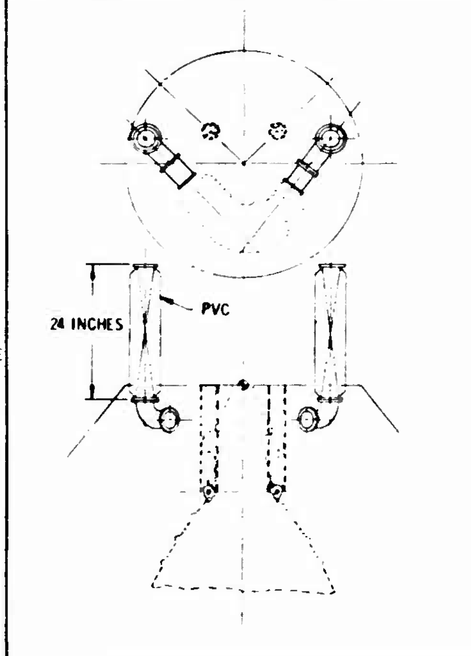
FUEL FLOWRATE - 7.61 LB/SEC

OXID FLOWRATE - 45.66 LB/SEC

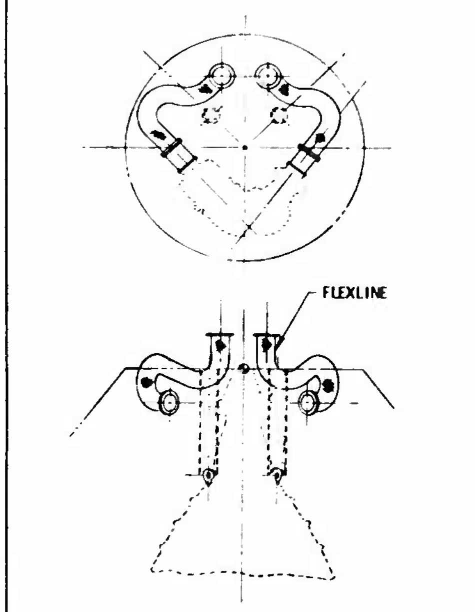
OPTION 1



OPTION 2



OPTION 3



DESCRIPTION	GIMBALLING SUCTION LINES		PVC SUCTION LINES		FLEXLINES (STAINLESS STEEL BRAID)	
PROPELLANT	FUEL	OXID	FUEL	OXID	FUEL	OXID
LINE DIA., IN.	3.0	3.0	3.0	3.0	3.0	3.0
VACUUM JACKET DIA., IN.	4.0	4.0	4.0	4.0	4.0	4.0
LINE LENGTH, IN.	44.0	57.5	33.00	34.00	33.0	34.00
LINE DRY WEIGHT, LB	10.99	13.20	14.20	14.37	11.59	11.89
TOTAL DRY WEIGHT, LB	24.19		28.57		23.48	
LINE WET WEIGHT, LB	11.76	29.85	14.36	17.27	12.17	21.74
TOTAL WET WEIGHT, LB	41.61		31.63		33.91	
NPSH INCREASE, PSI	0.96	3.03	1.06	2.29	4.90	10.71
R.O.M. COST	$9000		$16000		$6000	

Table XX. OOS Suction Line Configuration Matrix (Sheet 2 of 2)

TASK I
OOS SUCTION LINE CONFIGURATION MATRIX
FIXED BOOST PUMPS

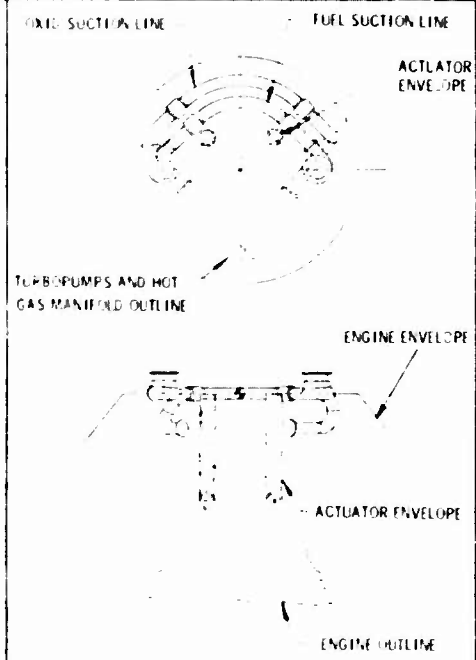
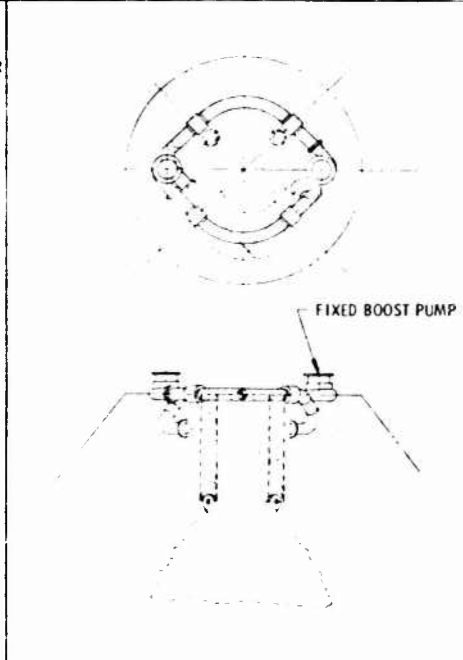
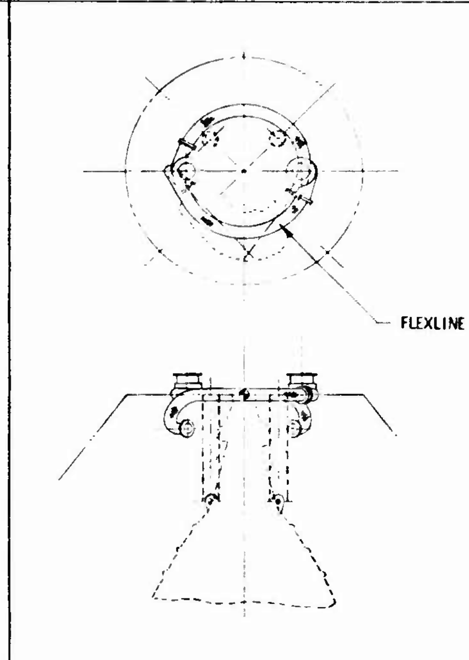
CONSIDERED CONFIGURATIONS	OPTION 1		OPTION 2		OPTION 3	
SPECIFICATIONS: GIMBAL ANGLE ± 5; FUEL FLOWRATE - 7.61 LB SEC; OXID FLOWRATE - 45.66 LB SEC						
DESCRIPTION	SUCTION LINES ON ONE SIDE		SUCTION LINES ON BOTH SIDES		FLEXLINES (STAINLESS STEEL BRAID)	
PROPELLANT	FUEL	OXID	FUEL	OXID	FUEL	OXID
LINE DIA. IN	2.00	2.00	2.00	2.00	2.00	2.00
VACUUM JACKET DIA. IN	3.00	3.00	3.00	3.00	3.00	3.00
LINE LENGTH, IN	43.5	55.0	43.5	38.5	33.0	37.0
LINE DRY WEIGHT, LBS (INCLUDING B.P. DRIVE LINES)	8.16	9.68	8.16	7.64	10.16	11.11
TOTAL DRY WEIGHT, LBS	17.84		15.80		21.27	
LINE WET WEIGHT, LBS	8.50	17.17	8.50	12.62	10.42	15.90
TOTAL WET WEIGHT, LBS	25.67		21.12		26.32	
NPSH INCREASE, PSI	0	0	0	0	0	0
R & M COST	$8000 + 3000 FOR B.P. DRIVE LINES		$8000 + 3000 FOR B.P. DRIVE LINES		$5000	

Table XXI. OOS Engine Design Configuration Matrix (Sheet 1 of 2)

TASK I

OOS ENGINE DESIGN CONFIGURATION MATRIX – STAGED COMBUSTION - PARALLEL TURBINE DRIVE

F = 25,000 lb, MR = 6.0, Nom. NPSH (F) = 60 ft, (OX) = 16 ft (Boost Pumps on all Configurations)

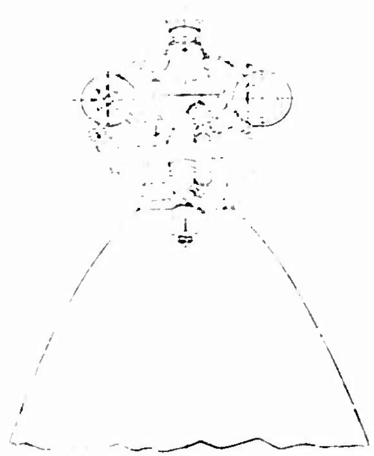
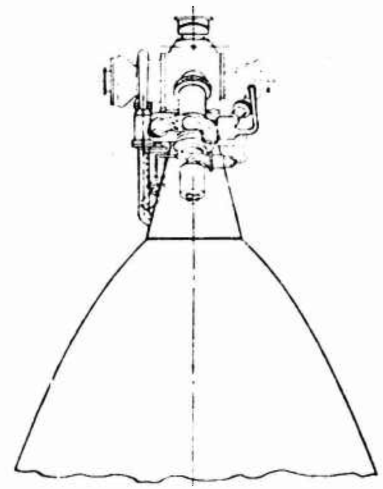
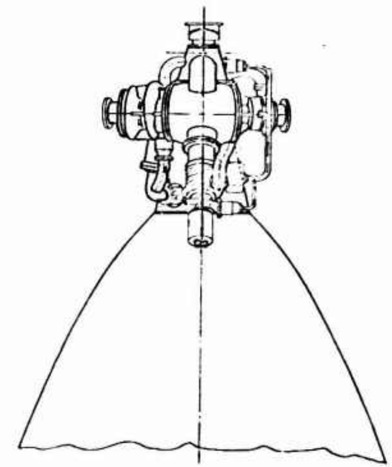
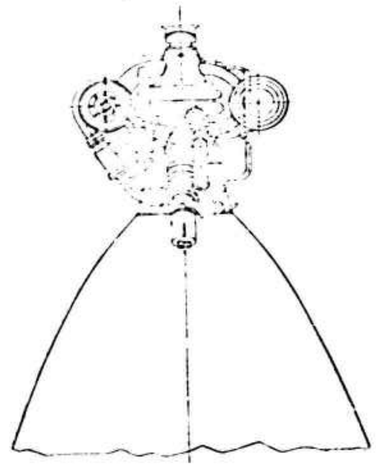
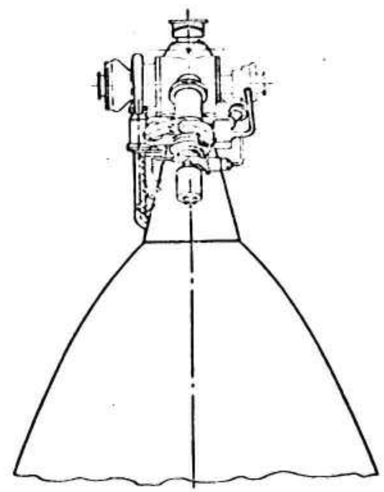
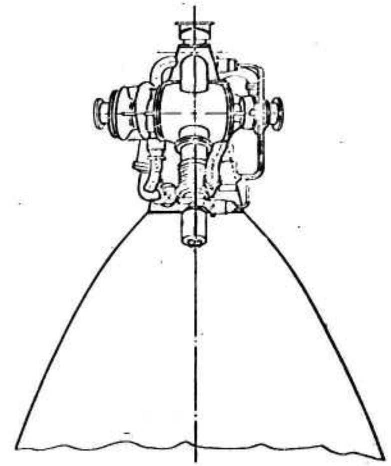
CONSIDERED CONFIGURATIONS			
DESCRIPTION	OPPOSED SIDE MOUNTED TPA'S, SINGLE PREBURNER, SEPARATE HOT GAS MANIFOLDS.	INLINE HORIZONTAL TPA'S (SEPARATE TURBINES) INJECTOR MOUNTED, COMMON PREBURNER AND HOT GAS MANIFOLDS.	HORIZONTAL SIDE MOUNTED "V" TPA'S, SEPARATE TURBINES, COMMON PREBURNER, HOT GAS MANIFOLD AND TURBINE EXHAUST RETURN MANIFOLD.
CONSIDERATIONS	OPTION A	OPTION B	OPTION C
[illegible]	[illegible]		
[illegible]	[illegible] RETRACTABLE.	⊖ [illegible] IN. LONGER ENGINE RESULTS IN ϵ_0 = 255:1, I_s = 1.1 W_{BO} DECREASE = 15 LBS, PL = 116 LBS (LOST).	⊖ SAME AS OPTION A.
[illegible]	[illegible]	⊕ SAME AS OPTION A.	⊖ SAME AS OPTION A.
[illegible]	[illegible] LBS, LOST BY MIN. WT. RETRACTABLE [illegible]	⊕ SAME AS OPTION A.	⊕ SAME AS OPTION A.
[illegible]	[illegible] THAN RETRACTABLE. W_{BO} = 51 LBS – FIXED NOZZLE (ϵ_0 = 390) VS MIN. WEIGHT RETRACTABLE (ϵ = 450) – RETRACTED TO 82 IN.		
[illegible]	[illegible]	⊕ INTEGRATED TURBINE MANIFOLD WITH INJECTOR HOT GAS MANIFOLD RESULTS IN LESS WEIGHT. ⊖ [illegible] REQUIREMENT MAY BE GREATER FOR LONGER ENGINE.	⊕ INTEGRATED TURBINE MANIFOLD RESULTS IN LESS WEIGHT THAN OPTION A.
[illegible]	[illegible] AREA [illegible]	⊕ INTEGRATED TURBINE MANIFOLD WITH INJECTOR HOT GAS MANIFOLD REQUIRE LESS FUEL CONSUMPTION IN SHUTDOWN LAG TO COOL ENGINE COMPONENTS.	⊕ INTEGRATED TURBINE MANIFOLDS RESULTS IN LESS FUEL CONSUMPTION IN SHUTDOWN LAG TO COOL ENGINE COMPONENTS THAN OPTION A.
[illegible]	[illegible]		
[illegible]	[illegible] TO ACCOMMODATE STOWAGE IN ORBITER PAYLOAD COMPARTMENT VERSUS FIXED NOZZLE CONFIGURATION.		
[illegible]	⊖ [illegible]	⊕ INTEGRATED TURBINE MANIFOLD WITH INJECTOR HOT GAS MANIFOLD REQUIRE LEAST TIME INTERVAL FOR SHUTDOWN AND SHORT COAST PRECHILL.	⊕ INTEGRATED TURBINE MANIFOLDS PROVIDES SHORTER TIME INTERVAL FOR SHUTDOWN AND SHORT COAST PRECHILL THAN OPTION A.

Table XXI. OOS Engine Design Configuration Matrix (Sheet 2 of 2)

TASK I

OOS ENGINE DESIGN CONFIGURATION MATRIX – STAGED COMBUSTION - PARALLEL TURBINE DRIVE

F = 25,000 lb, MR = 6.0, Nom. NPSH (F) = 60 ft., (OX) = 16 ft (Boost Pumps on all Configurations)

CONSIDERED CONFIGURATIONS			
DESCRIPTION	OPPOSED SIDE MOUNTED TPA'S, SINGLE PREBURNER, SEPARATE HOT GAS MANIFOLDS.	INLINE HORIZONTAL TPA'S (SEPARATE TURBINES) INJECTOR MOUNTED, COMMON PREBURNER AND HOT GAS MANIFOLDS.	HORIZONTAL SIDE MOUNTED "V" TPA'S, SEPARATE TURBINES, COMMON PREBURNER, HOT GAS MANIFOLD AND TURBINE EXHAUST RETURN MANIFOLD.
CONSIDERATIONS	OPTION A	OPTION B	OPTION C
III. MAINTENANCE CAPABILITY			
A. FIXED VERSUS RETRACTABLE NOZZLE	ALL OPTIONS: ADDITIONAL MAINTENANCE REQUIRED FOR RETRACTABLE NOZZLE MECHANISM AS COMPARED TO FIXED NOZZLE CONFIGURATIONS.		
B. ACCESSIBILITY TO ENGINE THROUGH [illegible] ENGINE ENVELOPE AND DESIGN FEATURES FOR	ALL OPTIONS: ENGINE AND COMPONENT ACCESSIBILITY BETTER ON ENGINES EQUIPPED WITH FIXED NOZZLES BECAUSE OF ABSENCE OF DEPLOYMENT GEAR REQUIRED ON ENGINE WITH RETRACTABLE NOZZLES.		
REMOVAL AND REPLACEMENT OF COMPLETE ENGINE	⊕ LOCATION OF TURBOPUMPS AWAY FROM THRUST TAKEOUT STRUCTURE TO VEHICLE AND OTHER INTERFACES MAY PROVIDE BETTER ACCESS.	⊖ FORWARD LOCATION OF TURBOMACHINERY MAY RESTRICT ACCESS.	⊖ SAME AS OPTION A.
IN PLACE REMOVAL AND REPLACEMENT OF LRU'S			
REMOVAL OF COMBUSTION CHAMBER (COMBUSTION CHAMBER)	⊖ ACCESS TO INJECTOR COMBUSTION CHAMBER MATING FLANGE MAY BE RESTRICTED BY INDIVIDUAL TURBINE MANIFOLDS AND PIPING.	⊕ BEST - TURBOMACHINERY POWERHEAD LOCATED FORWARD OF INJECTOR/COMBUSTION CHAMBER MATING FLANGE PERMITTING BETTER ACCESS.	⊖ ACCESS TO INJECTOR/COMBUSTION CHAMBER MATING FLANGE LIMITED IN ONE QUADRANT BY TURBOMACHINERY MOUNTING.
IN PLACE INSTALLATION AND [illegible] OF GIMBAL ACTUATORS TO ENGINE AND SPACE CRAFT.	ALL OPTIONS: DEPENDENT ON ENGINE VEHICLE ACTUATOR ATTACH LOCATION TO BE DEFINED THROUGH STRUCTURAL DYNAMICS ANALYSIS.		● APPEARS BEST BECAUSE TPA's UPDATED IN OPPOSITE QUADRANT TO PROPOSED ACTUATOR LOCATIONS.
[illegible] AND [illegible] IN PLACE REMOVAL AND REPLACEMENT	● TPA REMOVAL APPEARS FEASIBLE.	⊖ FORWARD LOCATION OF TPA'S MAY PREVENT REMOVAL FROM ENGINE BECAUSE OF ENGINE ENVELOPE RESTRICTION.	● SAME AS OPTION A.
VALVE IN PLACE REMOVAL AND REPLACEMENT	ALL OPTIONS: ELECTRICAL ACTUATION PERMITS ACTUATION MODULE REPLACEMENT WITHOUT CONTAMINATION OF PROPELLANT SYSTEM WHICH MIGHT BE EXPERIENCED IN COMPLETE VALVE REMOVAL.		
IV. DEVELOPMENT			
FIXED NOZZLE VERSUS RETRACTABLE	ALL OPTIONS: ADDITIONAL DEVELOPMENT EFFORT REQUIRED FOR RETRACTABLE NOZZLE AS COMPARED TO FIXED NOZZLE CONFIGURATION.		
THERMAL SHIELDING	⊖ THERMAL SHIELDING REQUIREMENT GREATER FOR SEPARATE TURBINE MANIFOLDS.	⊕ MINIMUM THERMAL SHIELDING REQUIRED FOR RADIATION TO OTHER COMPONENTS.	● THERMAL SHIELDING REQUIREMENTS LESS THAN OPTION A.
V. PRODUCTION COST			
FIXED NOZZLE VERSUS RETRACTABLE	ALL OPTIONS: PRODUCTION COST GREATER FOR RETRACTABLE NOZZLE AS COMPARED TO FIXED NOZZLE CONFIGURATIONS.		
VI. EVALUATION			
A. RETRACTABLE NOZZLE	3	2	1
B. FIXED NOZZLE	2	(1) LONGER ENGINE RESULTS IN LESS ϵ_e (230) AND RESULTANT PAYLOAD CAPABILITY LOSS (200 + LBS).	1

installation of PVC's or gimbaled suction lines. Although not shown on the engine assembly drawing, all small propellant recirculating lines and helium purge lines interface at the same station plane to keep a fixed flat engine vehicle hydraulic and electrical interface.

2 Electrical Systems

Location of the engine mounted controller at a proposed engine-vehicle interface station 4.50 in. aft of the gimbal point eliminates interconnect harnesses between the vehicle and interface support bracketry. Vehicle harnesses, with service loops for gimbaling, can then be directly connected to the engine mounted controller.

3 Gimbal System

a Gimbal Assembly

The gimbal assembly engine-vehicle interface mounting plane is 4.20 in. forward of the gimbal point. The design of the gimbal assembly is discussed in Section III,B,2,j.

b Gimbal Actuators

Gimbal Actuator mounting provisions are shown on Sheet 1 of the engine assembly drawing in Figure 127. Engine mounting clevices for pitch and yaw gimbal actuators are located on moment arms 9.2 in. from the engine centerline. The vehicle actuator attach point is shown on a plane through the gimbal point based on a calculated minimum actuator length. The attach point may be adjusted axially at the discretion of the vehicle contractor by extending the actuator ram arm length.

c Engine Payload Capability

A preliminary engine design included a retractable nozzle to obtain a maximum payload capability. The subsequent design replaced the retractable AGCarb nozzle extension with a fixed nozzle extension of the same material with an exit area ratio $\epsilon = 290:1$, based on the payload tradeoff analysis results shown in Table XXII. A payload loss of 198 lb payload was outweighed by reduced engine design complexity and cost of the fixed nozzle concept. Final baseline design was established as a full H_2 regeneratively cooled fixed nozzle. A negligible payload penalty of 14 lb was accepted for engine design improvement because the hot gas tubular nozzle-AGCarb nozzle extension flange was eliminated, as well as performance loss associated with cooling the joint.

TABLE XXII

ENGINE BASELINE EVOLUTION, ENGINE BURNOUT WEIGHT/PAYLOAD CAPABILITY COMPARISON

$F = 25000$ lb, $MR = 6.0$

Baseline Parameter	Initial Design Concept	Preliminary Design Concept	Interim Baseline	Final Baseline
P_c, PSIA (Life Factor)	1500	1800	1800	1800
Exit Area Ratio, ϵ_o	450	450	290	290
Nozzle Configuration	Min Wt. Retractable AGCarb Nozzle Extension	Min Wt. Retractable AGCarb Nozzle Extension	Fixed AGCarb Nozzle Extension	Fixed Full Regen. Nozzle
Transition ϵ	145	125	125	N/A
Engine Burnout Wt. W_{BO}, lb	544.6	512.3	417.0	459.3
ΔW_{BO} vs Baseline	+85.3	+53.0	-42.3	---
I_s, Sec	468.4	469.1	465.7	465.7
I_s, Sec (-0.9 Sec for joint cooling)	467.5	468.2	464.8	465.7
ΔI_s vs Baseline	+1.8	+2.5	-0.9	---
ΔPL, lb vs Baseline (1)	-31	+198 (2)	+14 (3)	---

NOTES: (1) $\Delta PL = 157\ \Delta I_s - 3.68\ \Delta W_{BO}$

(2) Payload gain of 198 lb outweighed by a reduced engine design complexity obtained with baseline fixed nozzle concept.

(3) Negligible 14 lb payload gain considered outweighed by full regenerative nozzle baseline concept where hot gas transition flange and cooling loss prediction is eliminated

(3) Baseline Engine Configuration Evolution

Parametric analyses established initial engine cycle and design point characteristics for the 25,000 lb thrust staged combustion engine. Evolution of the final baseline engine configuration through selected interim baseline mileposts is presented as a flow diagram in Figure 129 and discussed in chronological order in the following discussions:

(a) Initial Design Concept

Engine cycle analyses produced the staged combustion single preburner cycle as the preferred candidate cycle for baseline. The staged combustion-parallel turbine drive received primary consideration with the staged combustion bleed-series turbine variation also under active consideration, as shown in Figure 129. Chamber pressure was selected at 1500 psia for both cycles as was a preburner-turbine inlet temperature of 1860°R. Engine configurations were established and are presented for the staged combustion-parallel turbine drive and staged combustion bleed cycles in Figures 130 and 131, respectively. Both engine configurations used retractable AGCarb nozzle extensions with an exit area ratio ε = 450:1 for maximum payload capability. Powerhead configurations were similar with both engines utilizing opposed side mounted TPA's in separate turbine manifolds. Boost pumps were engine mounted. Gimbaled crossover suction lines were utilized on both engines.

(b) Preliminary Design Concept

Chamber pressure was increased from 1500 psia to 1800 psia based on thermal life analysis.

The staged combustion parallel turbine drive cycle was selected as baseline because of maximum performance as shown in Figure 129. The engine configuration retained the retractable AGCarb nozzle extension concept at ε = 450:1. Based on the selection matrix results in Table XXI, the baseline power head was established with turbopumps side mounted laterally in a "V" about the engine to an integrated turbine housing, as was the single preburner. Two engine configurations were designed reflecting the above criteria. One engine design featured engine mounted in-line boost pumps integrated with the high pressure turbopumps (HPTPA's) as shown in Figure 132. The second engine featured remote vehicle mounted boost pumps as shown in Figure 133.

(c) Interim Baseline

Subsequent to the 90-day data dump, baseline suction line routing options were defined through the selection matrix

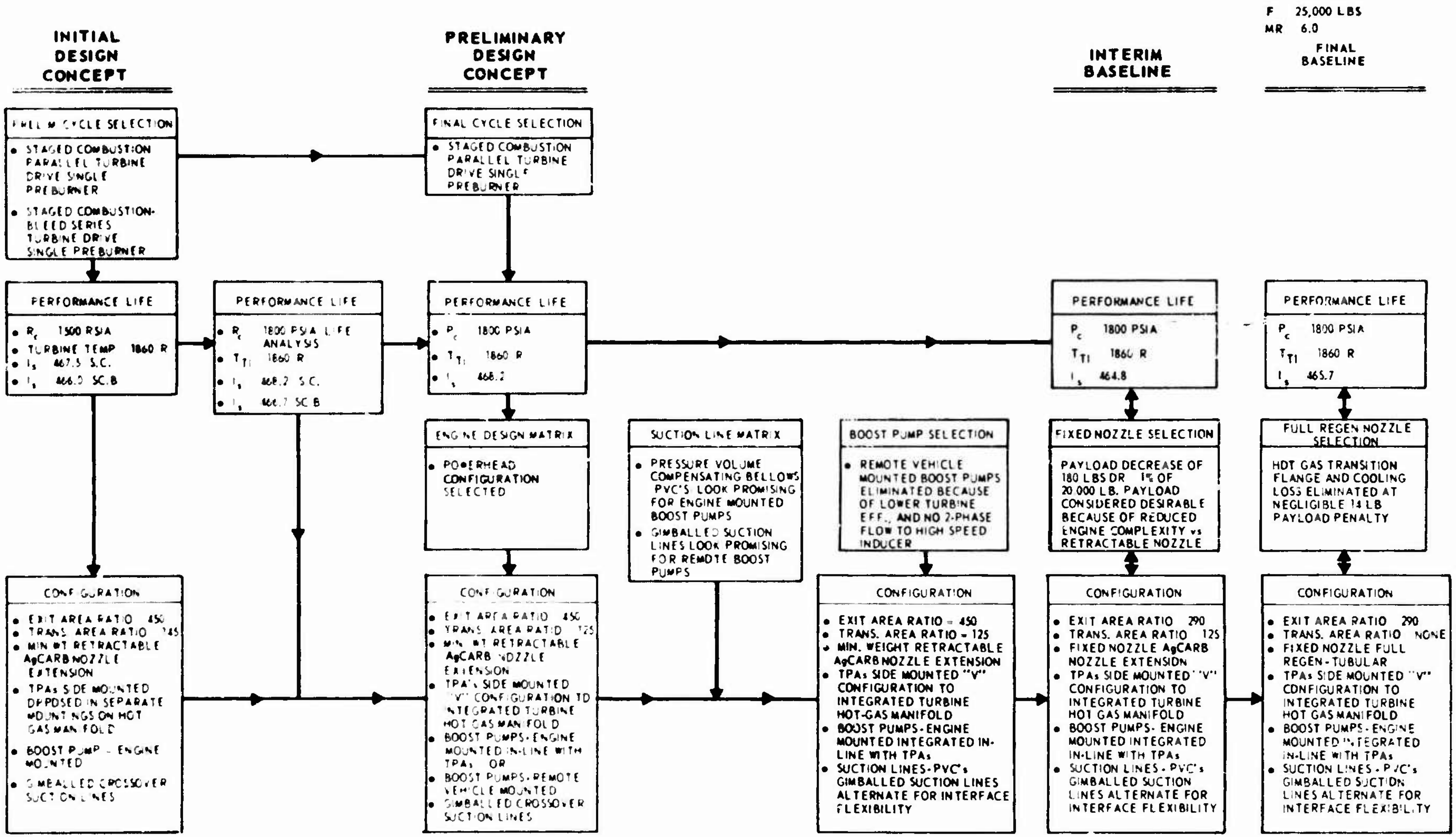


Figure 129. Baseline Engine Configuration Evolution

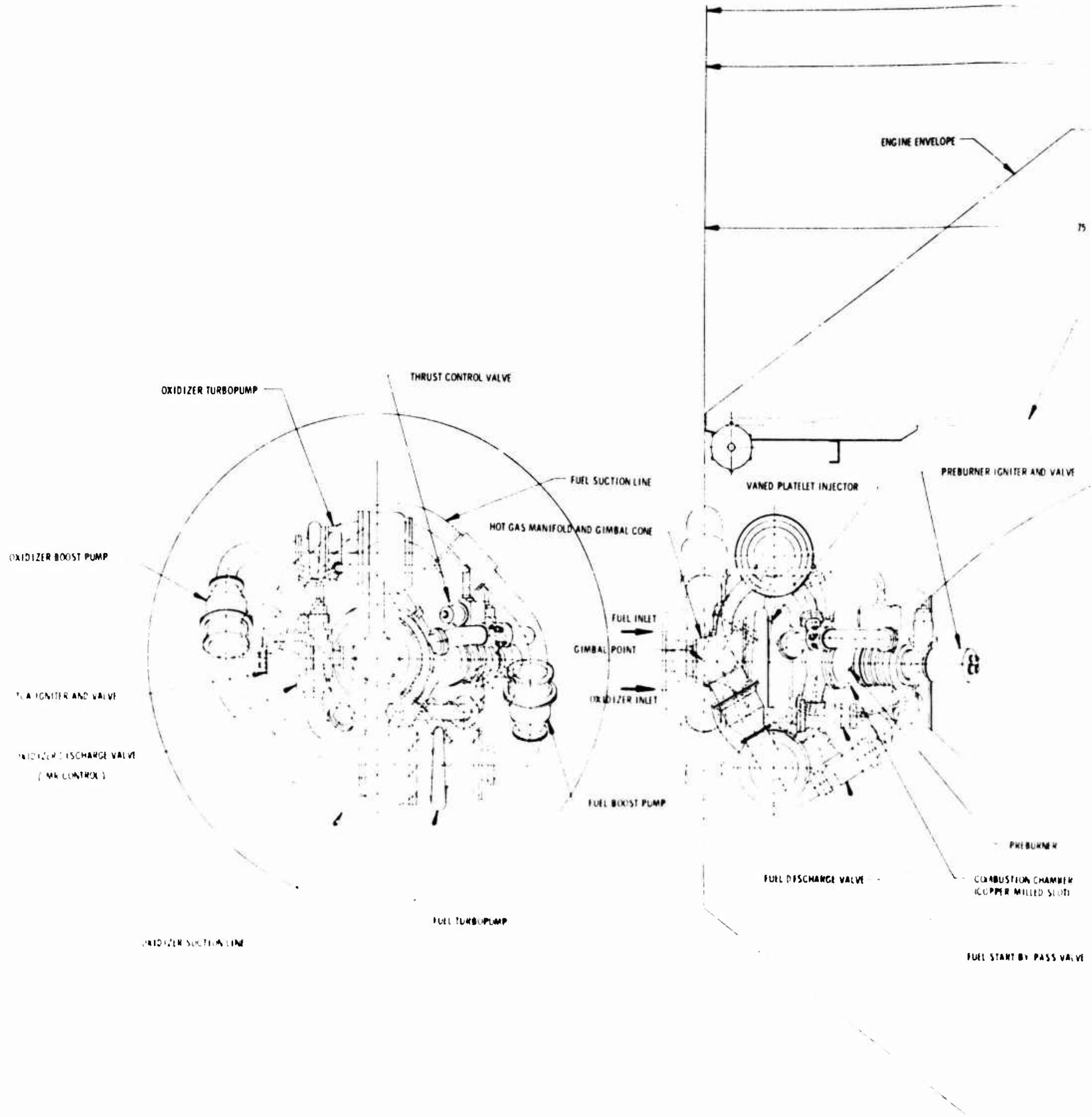
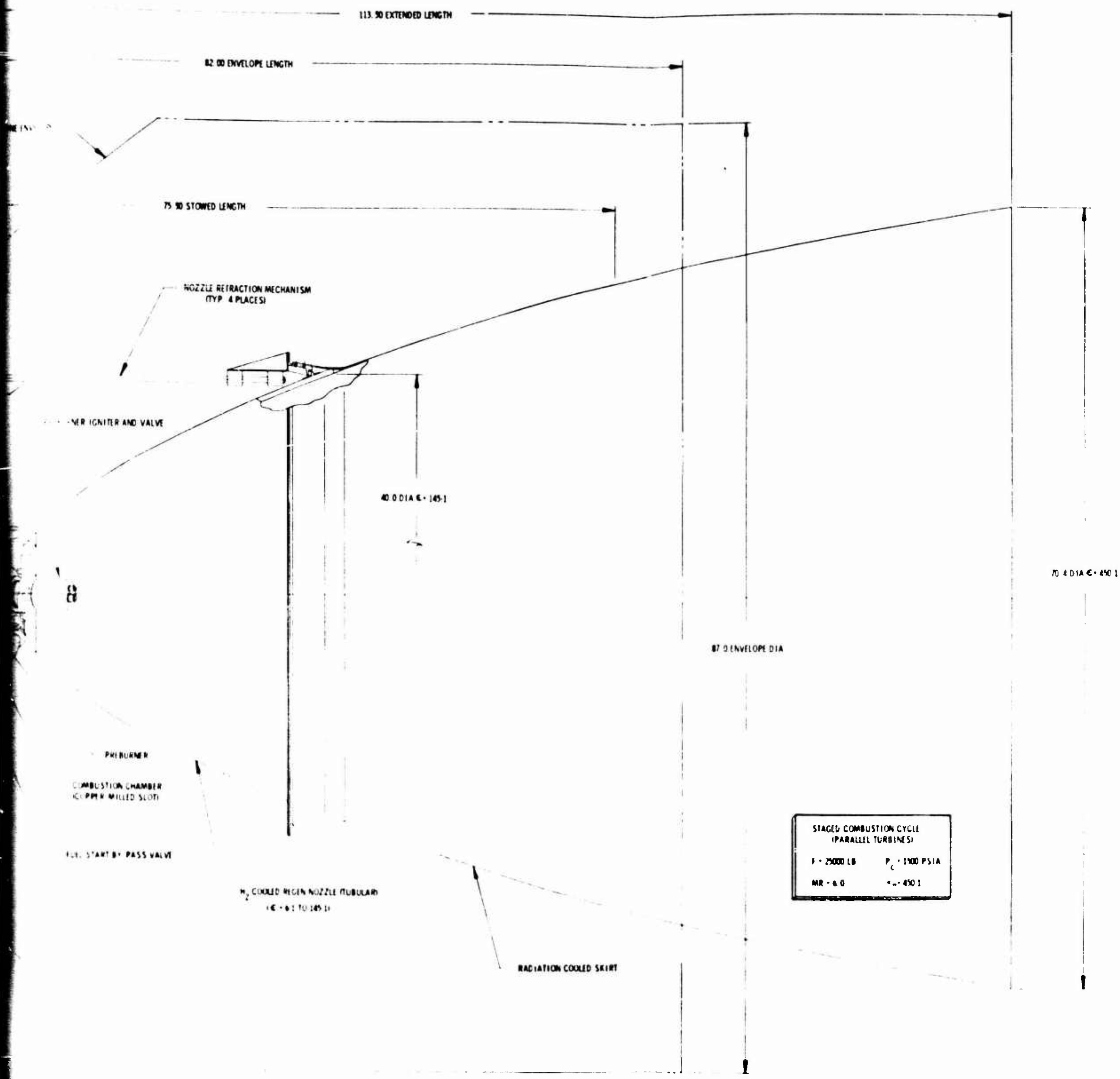


Figure 130. Staged Combustic
Engine



...Combustion Cycle, Parallel Turbine Drive

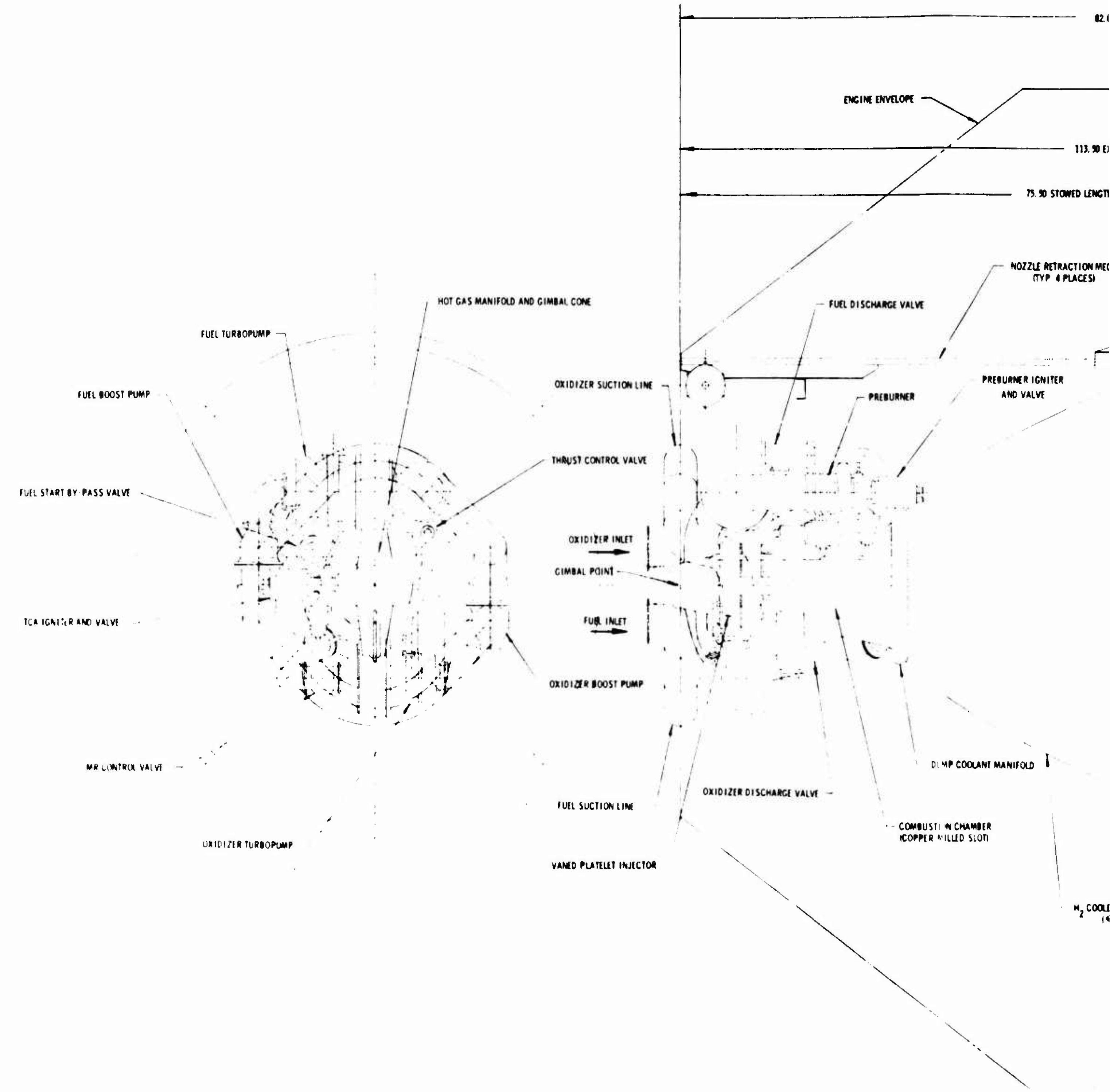
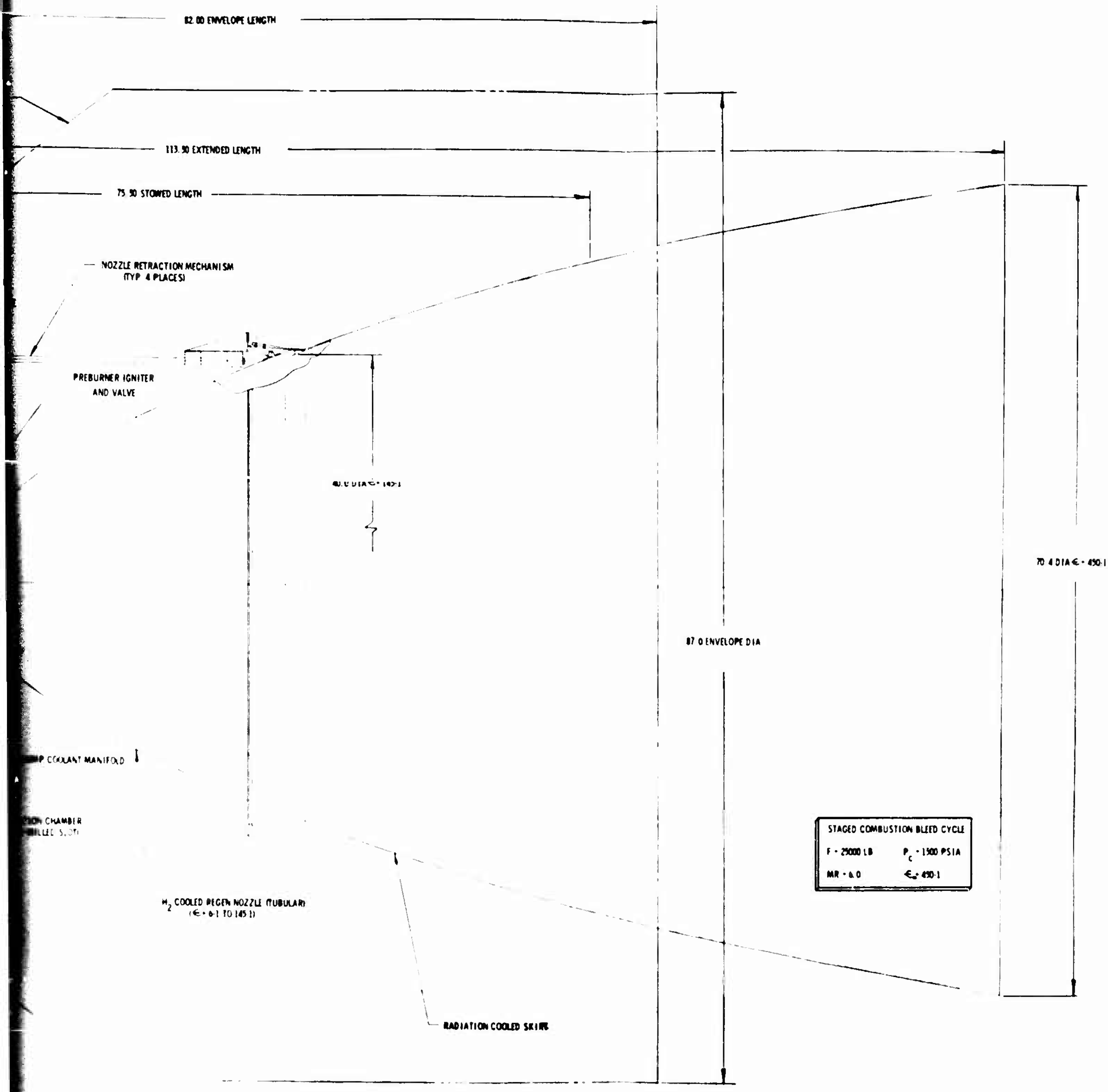


Figure 131. Staged Combustion
Engine



Combustion Bleed Cycle, Parallel Turbine Drive

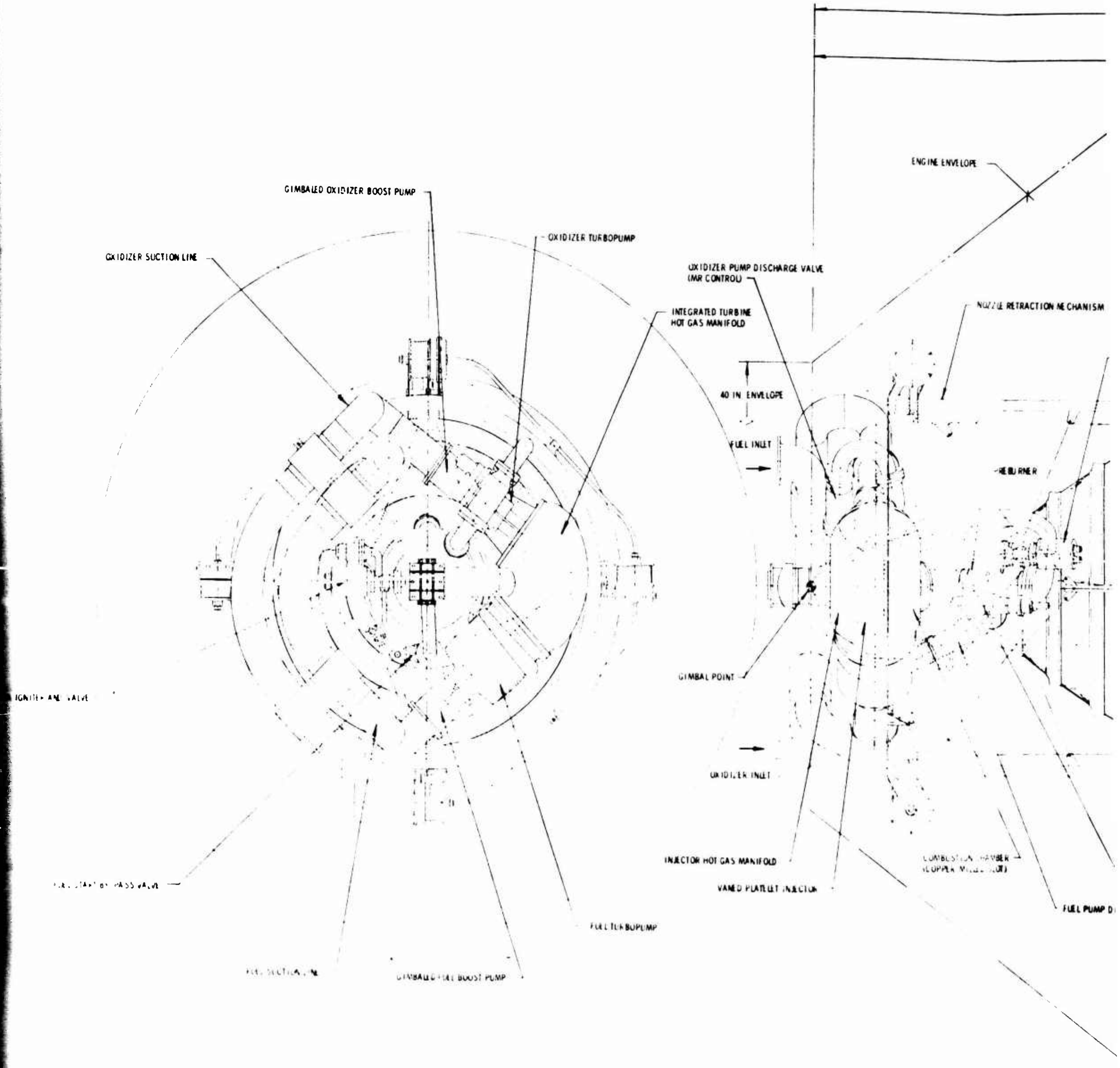
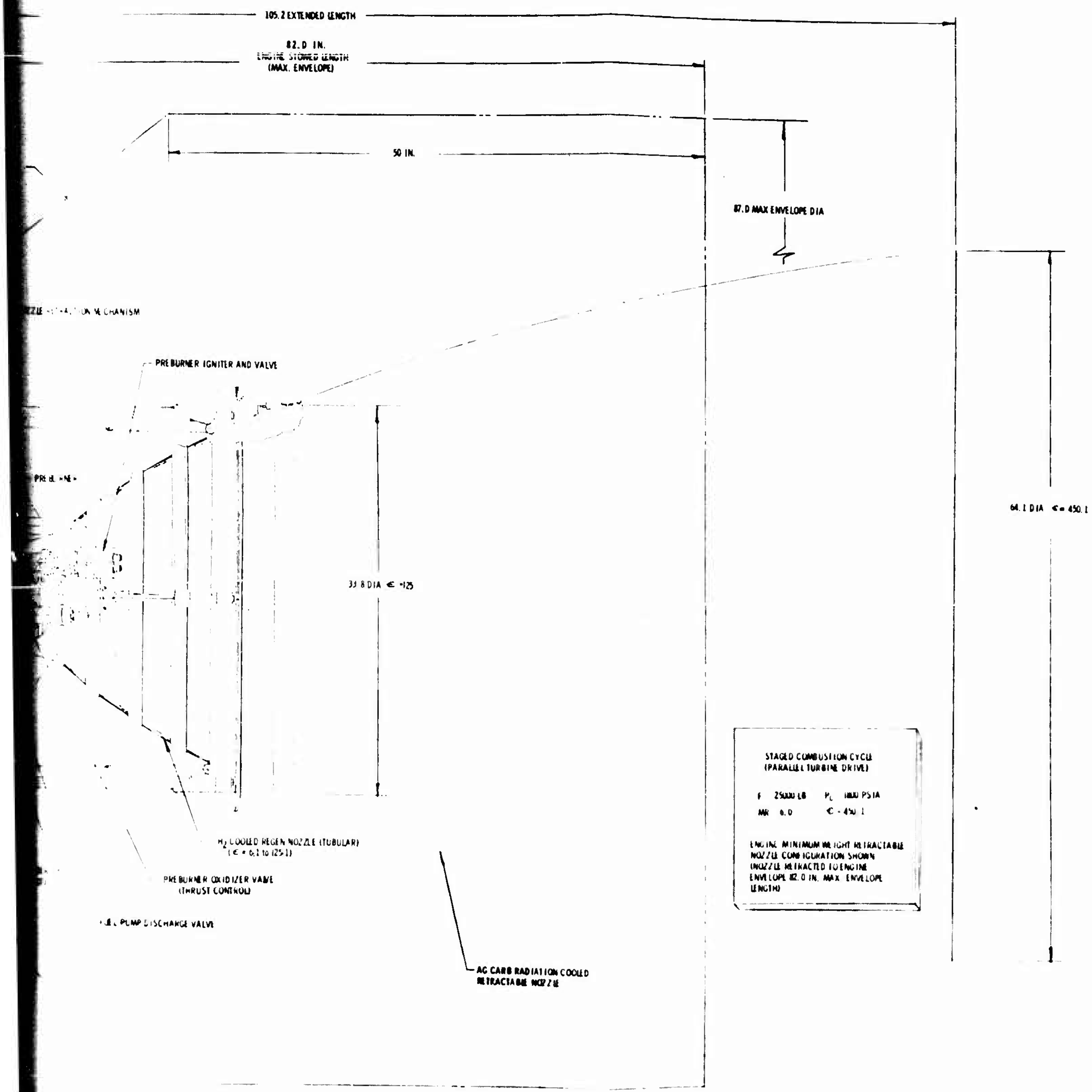


Figure 132. Staged Combustion Cy
with Integrated Boos



... Cycle Engine, Parallel Turbine Drive
... Boost Pumps

2

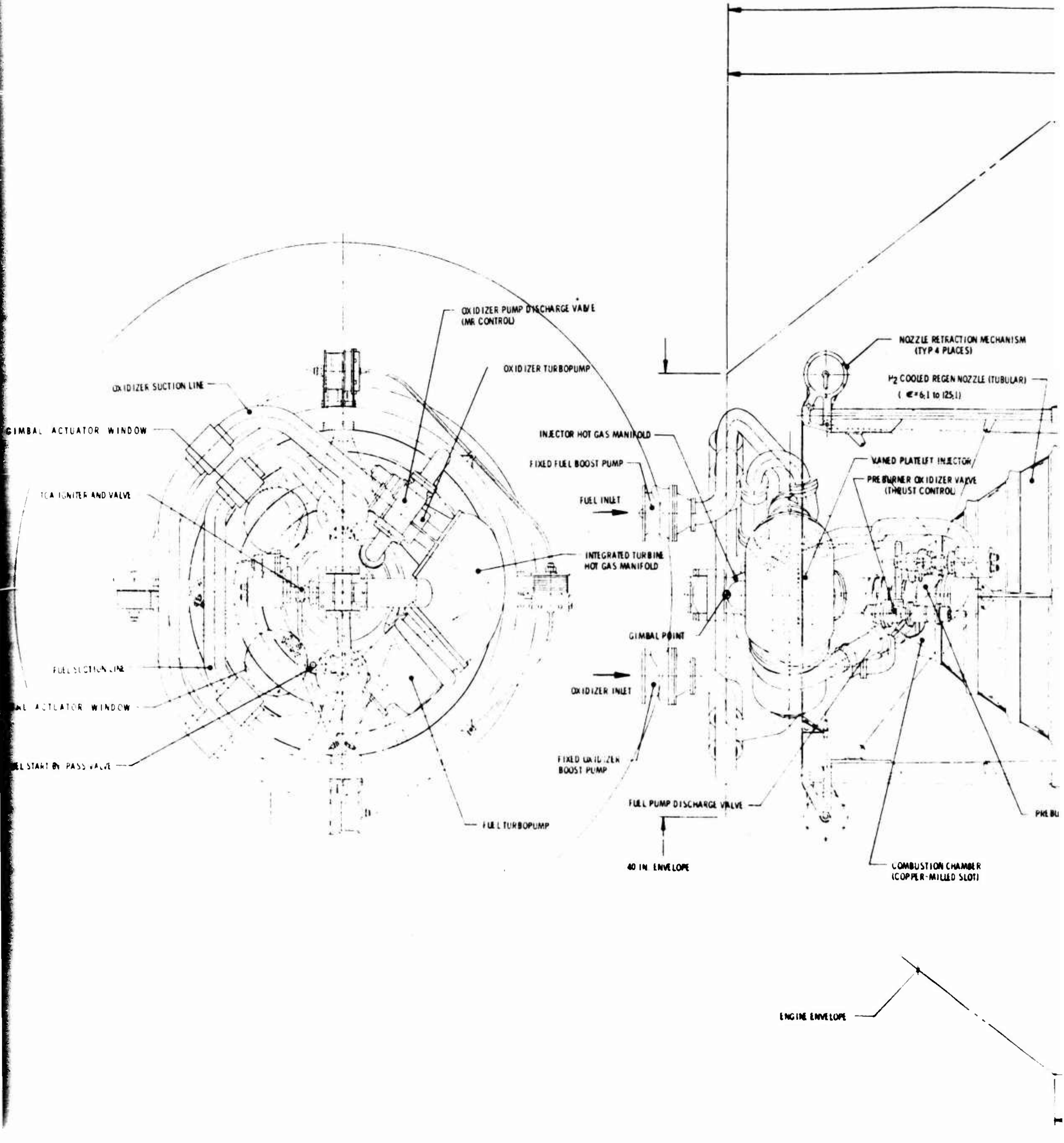
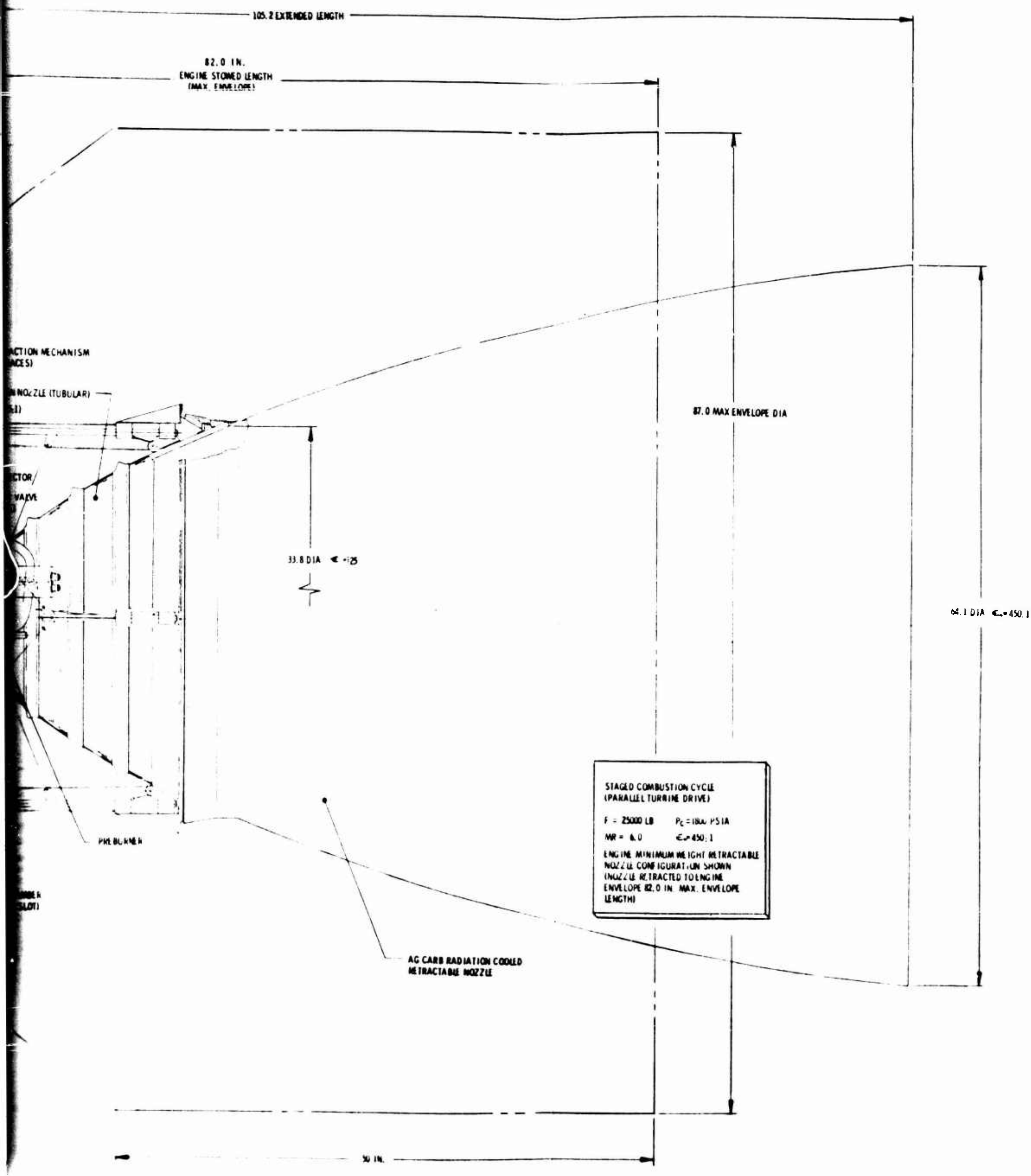


Figure 133. Staged Combustion Cy
with Remote Boost Pu



Cycle Engine, Parallel Turbine Drive
Boost Pumps

presented in Table XX. Boost pumps were baselined as engine mounted in-line integrated units with the high pressure turbopumps (HPTPA's). Selection of engine mounted boost pumps resulted in baseline definition of suction line routing to vaned elbows attached to the boost pump inlets with adaptability to employ either pressure volume compensating bellows (PVC's) or gimbaled suction lines to create interface flexibility for vehicle contractor suction line routing selection.

The engine nozzle configuration was changed from a retractable AGCarb nozzle with an exit area ratio ε_o = 450:1 to a fixed AGCarb nozzle extension with an exit area ratio ε_o = 290:1 because of the simplified engine design and reduced cost at a resulting payload loss of <1°, as shown in Table XXII. Engine configuration layouts were in process reflecting the above design changes but an engine assembly drawing was delayed for final baseline.

(d) Final Baseline

The engine nozzle configuration was changed from a fixed AGCarb nozzle extension from ε = 125:1 to 290:1 to a full H_2 regeneratively-cooled tubular fixed nozzle configuration to ε_o = 290:1. This design change eliminated the hot gas transition flange joint between the tubular nozzle and AGCarb nozzle extension as well as associated flange cooling requirements.

Final baseline engine assembly Drawing 1161635 presented in this report as Figure 127, was completed and included all design changes subsequent to the 90-day data dump discussed in Section (d) above and the final full H_2 regeneratively-cooled fixed nozzle configuration discussed above.

(4) Structural Design Criteria

Structural design criteria were established for the OOS Engine in conformance with the contract. Appendix C presents the structural design criteria, loading definition and failure modes analysis.

(5) Materials Definition

Principle materials utilized on the OOS Engine are defined on engine assembly drawing 1161635, Sheet 2, (Ref. Figure 127). Detailed material callouts are found on the component drawings and within the text describing each component design in this report.

e. Engine Chilldown Methods

(1) General

Four basic systems for chilling and starting the OOS engine were investigated in determining the most suitable. It is

extremely difficult to simulate space environment childown conditions. Consequently, systems requiring elaborate childown and start cycles were considered as well as rather simple systems to ensure conservatism for initial test. It will be noted that all of the systems are similar and that no definite recommendations are made with respect to the preferred system. This is because the final selection must be predicated upon the characteristics of the actual hardware and mission. Consequently, the systems described in this section should be considered representative of the type which will evolve as a result of the detail design effort.

The principle childown method currently selected for the OOS engine is sequentially propellant dumping through TCA. An idle mode in conjunction with the childown was evaluated as an extension of this system and is discussed in Section III,B,1,f. Several other methods are also discussed as options, and of these the recirculation method holds considerable promise. Recirculation enables stable and repeatable starts, better performance, and may ultimately simplify system requirements and complexity.

The sequences presented were prepared assuming the engine had reached equilibrium ambient space environment temperature. However, in many instances the engine will be restarted after various coast periods. These variations in the engine prestart condition are compensated for by having the childown and start sequences controlled by an open loop system which senses the TPA temperatures, suction pressure, suction temperature and line pressures. Based on these parameters the sequences can be initiated and proceed from any point, depending on the mission, to precondition the propellant feed systems and stably start the engine.

All four systems described below require a propellant settling mode and assume settling is provided, to at least 0.1g, by firing the APS motors. In the event the APS motors cannot be utilized for settling, an alternate settling mode was considered. Propellant settling, by the alternate mode, is accomplished by opening the fuel discharge valve and igniting the thrust chamber igniter. The fuel will vaporize and expand through the thrust chamber nozzle providing the required settling impulse.

(2) Bleed Childown

Childown experiments conducted during the Saturn S-IVB flights SA-504 and -505 showed an adequate system childown may be obtained by merely sequence bleeding some propellants through the engine. An engine schematic of the propellant bleed childown system is presented in Figure 134. This system has the advantage of being simple with high component reliability. Its primary disadvantage is mission performance loss. Based on Saturn S-IVB data and data prepared for the space shuttle engine (SSE) system, it is anticipated that an average mission I_{sp} loss of 1.0 sec may result.

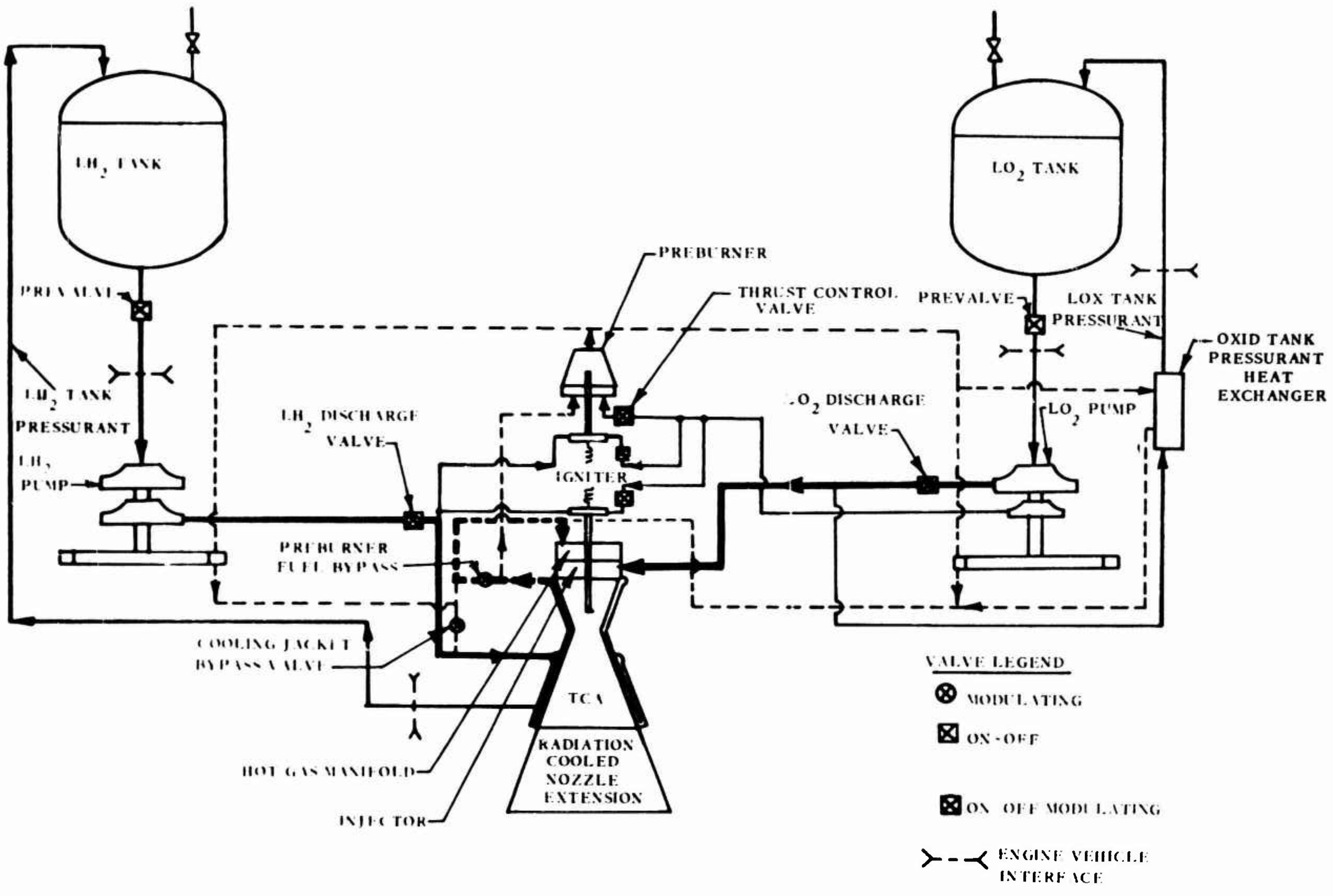


Figure 134. Bleed Chilldown System

No tank pressurization is used during the childown. The propellants, merely vented through the engine provide sufficient flow to chill the engine. More rapid childown can be accomplished using helium to pressurize the tank and consequently increase flow rates.

(a) Childown

When the propellants are sufficiently settled, the oxidizer discharge valve is partially opened allowing oxidizer to flow through the first stage of the oxidizer pump and exhaust through the thrust chamber. This flow is continued until the propellant quality at the oxidizer inlet is adequate for a stable engine start. The oxidizer valve is then closed and a period is allowed to evacuate the thrust chamber. The preburner fuel bypass valve is opened and the fuel discharge valve is then partially opened to allow fuel to exhaust through the thrust chamber. Flow through the cooling jacket is controlled by the cooling jacket bypass valve and is limited to just enough flow to chill the thrust chamber tubes to ensure a stable fuel flow during the early part of the start transient.

When the quality of the fuel at the pump suction is zero NPSH, the engine ignition and start mode is initiated.

An option to this system which will be considered, is to include recirculation lines and valves (as shown in Figure 135) from the fuel pump discharge and the first stage oxidizer pump discharge to the respective tanks. With these recirculation circuits, propellants may be permitted to remain trapped between the tankage prevalves and the discharge valves to maintain a chilled condition during vehicle coast periods. As the system warms up during the post shutdown coast periods, vapors formed in the pump systems will bleed back to the tanks.

In the event the APS motors cannot be utilized for settling the propellants, fuel will be exhausted through the thrust chamber by opening the fuel discharge valve and positioning the cooling jacket bypass valve so the chamber coolant tubes are bypassed. The impulse provided by this flow will settle the propellants.

(3) Optional Systems

(a) Propellant Tanks Vented Through Engine

This system has two modes of operation; coast chilling and prestart chilling. The only difference between the previously discussed, bleed chilling system, and this system is the prechill during coast. To accomplish the prechill the tank vents are plumbed into the respective TPA suction lines through a two-way valve, see Figure 136. A vent downstream of the pump, discharges through a discharge vent valve. During the coast periods propellant vented through the TPA will provide a continuing chilling media. The amount of additional chill required prior to starting the

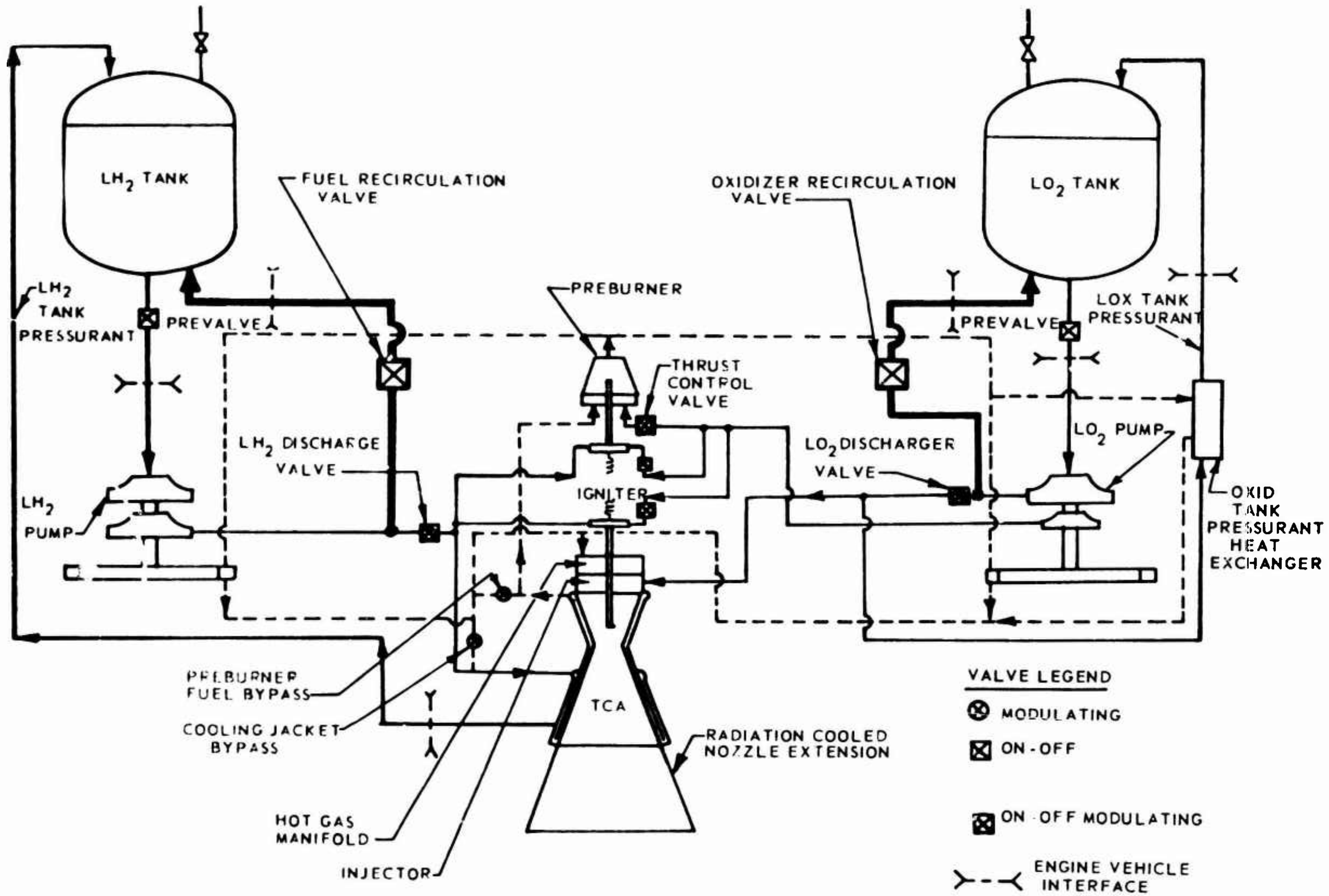


Figure 135. Bleed Childown System Option

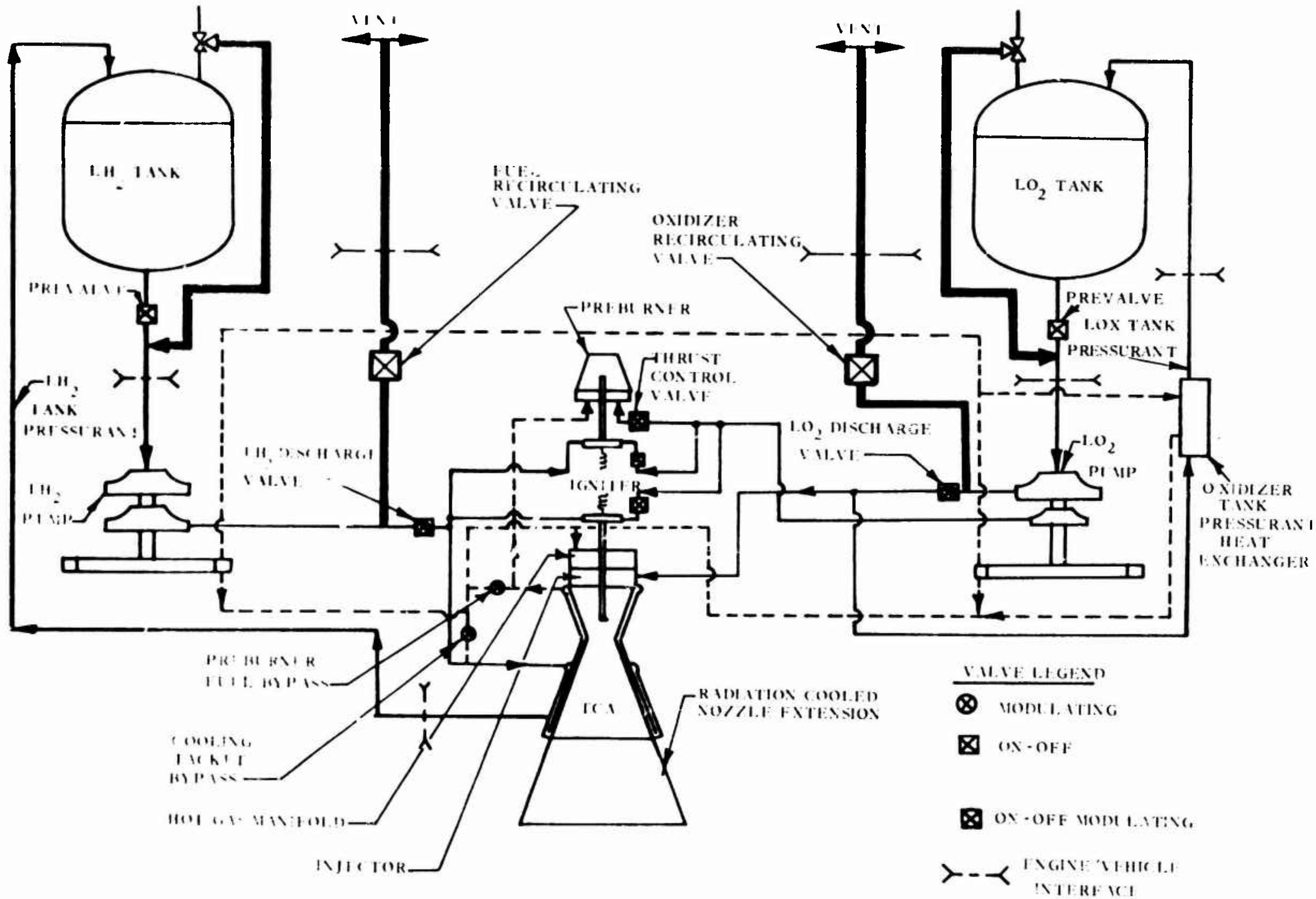


Figure 136. Propellant Tanks Vented Through Engine Childown System

engine will depend on the duration of the coast and the number of previous engine operations. The control system will sense the engine condition prior to a start and proceed with the settling, engine chilldown, ignition, and start sequence.

The sequence for chilling the engine with this method is the same as described for the bleed system except the tank vent valves will be actuated from the engine position to the atmospheric position and the pump discharge vent valves are closed prior to proceeding with the sequence.

(b) Wet Engine

The wet engine system requires propellants loaded with the prevalves opened and the TPA's are chilled as the loading proceeds. Through the entire mission the prevalves are then kept open, being closed only in the event of a failure or leaking discharge valves. Recirculation lines to the tank with valves are required to prevent trapped vapors during the settling mode. No chilldown sequence prior to start would be required for this system. Prior to start a settling mode as described earlier would be initiated. A major difference however is that the oxidizer propellant system up to the preburner valve would most likely be chilled and the oxidizer would be in a liquid state. Hence, a dual manifold preburner injector may be required, one to control preburner mixture ratio during the start, at which time the oxidizer is a liquid, and another for steady-state operation when there is enough heat to vaporize the oxidizer as it passes through the preburner cooling jacket.

Figure 135 is a system schematic showing the tank return lines and valve necessary to eliminate any trapped vapors during the settling mode.

(c) Turbopump Recirculation

After the propellants have been settled by either the APS or fuel bleed they are circulated from pump suction to tank by convective flow (see Figure 137). Heaters in the recirculation circuit provide sufficient energy to maintain reasonable recirculation flow for a rapid chilldown. During the chilldown, propellants are kept settled by firing the APS or flowing a small amount of fuel through the chamber.

Options to this system are the use of pumps to recirculate the propellants. A separate pumping system can be used in the recirculation circuits or the boost pumps may be rotated by auxiliary motors to act as recirculation pumps. All these systems, however, effect the engine design criteria and require further evaluation. Power requirements, engine operation, weight, reliability, and maintainability must all be considered in the evolution of the optimum system.

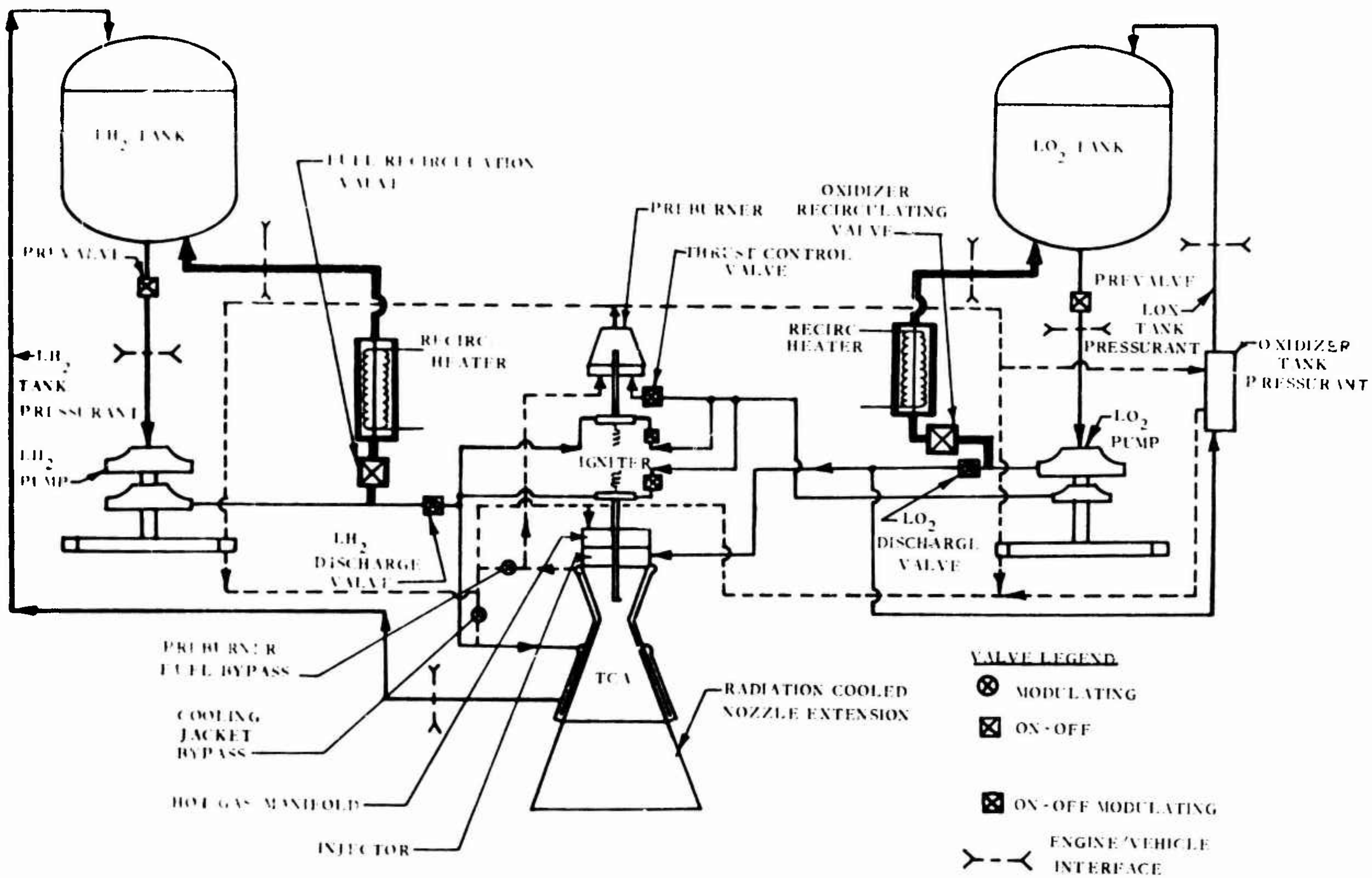


Figure 137. Turbopump Recirculation Chilldown System

f. Engine Idle Mode

With some modifications to the basic engine design, the OOS Engine is capable of operating in an idle mode. The idle mode, if required, will provide propellant settling impulse, usable impulse during the bleed childown cycle, and vernier thrust levels for small delta-V corrections. An investigation was conducted to determine the most suitable system for the OOS Engine. Table XXIII shows the results of the investigation and selection made and Figure 138 shows a schematic of the system. The system selected has two phases; the chilldown phase and the post shutdown phase.

Table XXIV presents the idle mode operation. Column 1, in the Table, presents the idle mode performance at the initiation of the idle mode with the pump not chilled at zero NPSH. Column 2 presents the performance after the pump is chilled (for both 0 and tank pressure NPSH) the combustion chamber has reached thermal equilibrium, and the preburner igniters are operating and ready for the initiation of full engine thrust operation. The engine may be ramped to full thrust any time after the pump has been chilled and the tanks pressurized. The idle mode data shown in Column 2 are also representative of the engine performance during the post shutdown idle mode. Column 3 presents the idle mode engine operating conditions at nominal tank pressure and pumped chilled, indicating the idle mode thrust and mixture ratio sensitivity to tank pressure conditions.

Figure 139 presents the fuel pump chilldown parameters with the cooling jacket bypass valve position varying to maintain a constant thrust chamber mixture ratio during the chilldown period. For each of these conditions the respective thrust chamber mixture ratio, thrust, and the fuel flow rate are plotted. The significance of the data is that if desired a more complex control system can provide a higher overall chilldown idle mode performance.

(1) Pressure-Fed Chilldown Idle Mode

Prior to starting the chilldown idle mode, the preburner bypass valve and the cooling jacket bypass valve are opened. The fuel discharge valve is then opened and the thrust chamber igniter spark plug is energized. Some fuel flows through the cooling jacket; however, most of the fuel bypasses the cooling jacket and flows directly to the thrust chamber injector and is expanded through the thrust chamber providing initial settling impulse. When the propellants are sufficiently settled the thrust chamber igniter oxidizer valve is opened and the igniter is ignited. The oxidizer discharge valve is then opened and the thrust chamber is ignited. Upon sensing ignition, the igniter is de-energized and the igniter oxidizer valve is closed.

To provide tank pressure prior to engine start, helium gas stored at liquid hydrogen temperatures could be heated and impelled into the tanks as shown in the engine schematic, Figure 138. The helium could be heated by the use of a cooling jacket around the thrust chamber igniter. With this system, the thrust chamber igniter is operated during the entire idle mode.

The engine is ramped to full thrust, any time after the engine system is adequately chilled, by igniting the preburner igniter,

TABLE XXIII

IDLE MODE COMPARISON

SYSTEMS	ADDED ENGINE REQUIREMENTS	COMMENTS
1. Chilldown		
A. Single Propellant		
1. Oxid Bleed	None	Not practical because of high mission performance loss
2. Fuel Bleed	Modulating FDV	Not practical because of high mission performance loss
B. Pressure Fed Bipropellant		
1. T.C. Igniters	Modulating T.C. igniter fuel valve Fuel line from pump discharge to T.C. igniter Modulating T.C. igniter valve T.C.igniter redesign with cooled chamber closed loop igniter M.R. control	Additional control required Extensive system alteration Very low thrust
2. T.C. Igniter with T.C. fuel flow	Modulating FDV Modualting T.C. igniter oxid. vlave T.C. igniter redesign with cooled chamber Closed loop igniter M.R. control	Low Performance Complex control required for igniter MR Extensive base system alteration
*3. T.C. Igniter with fuel & Oxid chamber flow	Modulating FDV Modualating T.C. igniter oxid. valve T.C. igniter redesign with cooled chamber Closed loop igniter M.R. control	Better Performance Additional control required
4. T.C. Fuel & Oxid Flow with Igniter Off after ignition	Modulating FDV Helium pressurization system	Better Performance Some base system redesign Additional control required
C. Pump Fed		
Fuel & Oxid Flow with igniters off after ignition	Modulating T.C. igniter oxid. valve Modulating FDV Closed loop T.C. M.R. control	Requires prechill of TPA's & may be used for post shutdown idle Best Performance Additional control required

* Selected System

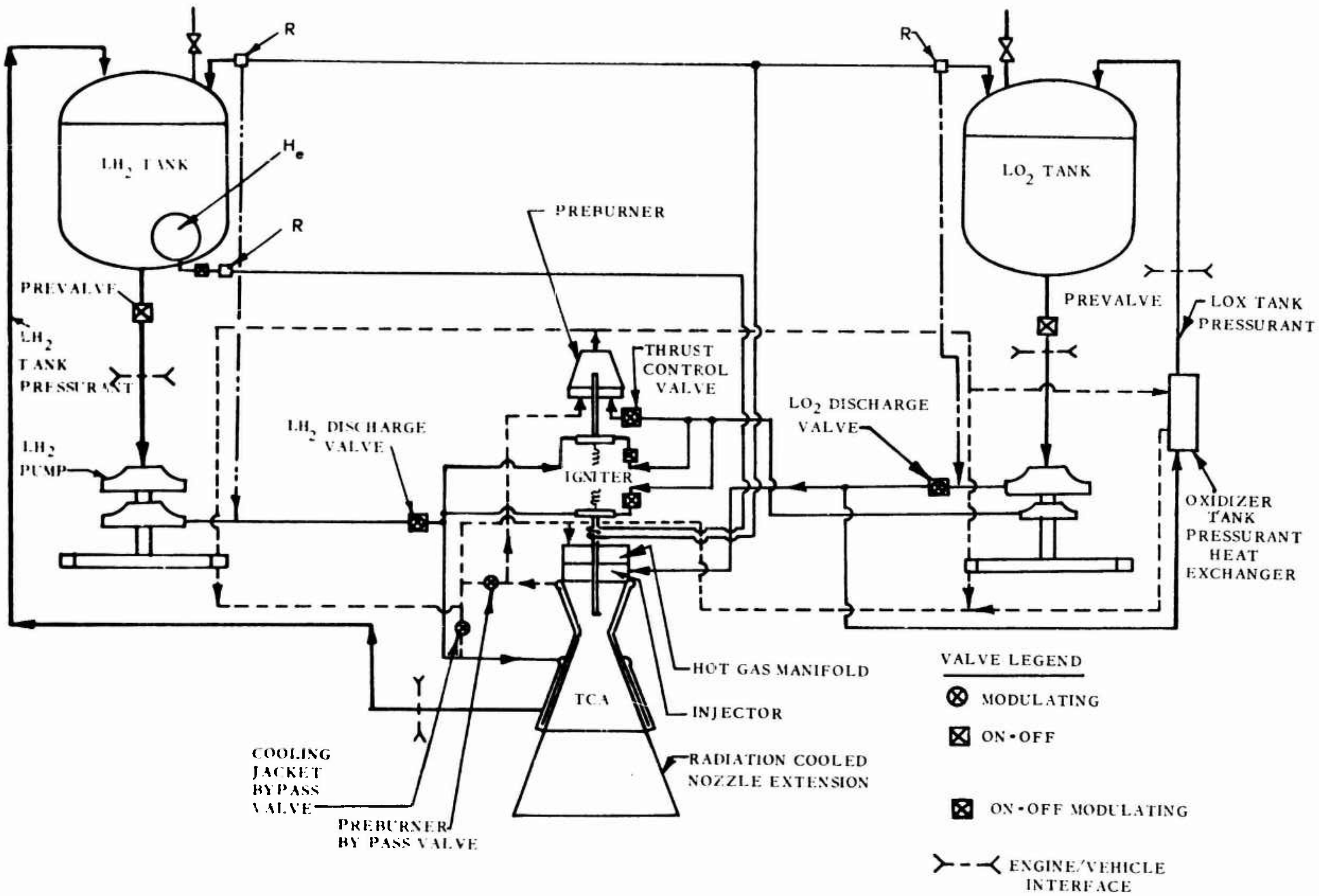


Figure 138. Idle Mode Schematic

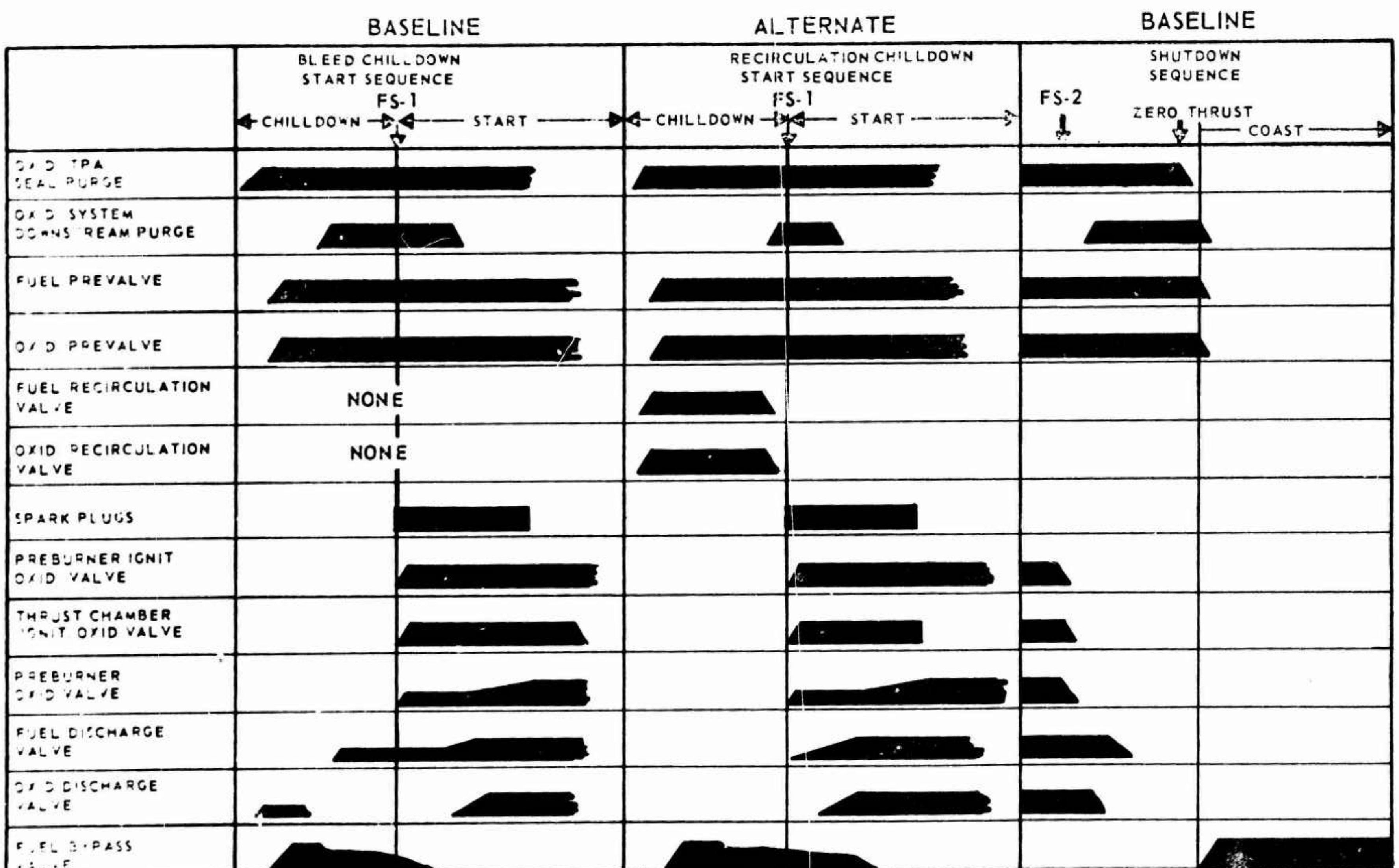
TABLE XXIV

PRESSURE-FED IDLE MODE OPERATION

		Initial Idle Mode Performance Prior to Pump Childdown		Idle Mode Performance at Thermal Equilibrium just Prior to Engine Start
Preburner Igniter		OFF	ON	ON
NPSH		0	Zero (ONPSH)	Tank Pressurized
Fuel Enthalpy - Pump Discharge	Btu	1800	-110	-110
Thrust	lb	59	57	74
Mixture Ratio		1.86	2.56	3.02
Specific Impulse	sec	412	431	442
Ox T.C. Valve % K_w		1.6	1.6	1.6
Fuel PB Bypass Valve % K_w		100.	100	100
CJKT Bypass Valve % K_w		100.	0	0
Fuel Tank Pressure	psi	15.	15.	17
Preburner Pressure	psi	5.7	5.5	6.6
Turbine Exhaust Pressure	psi	5.5	5.1	6.4
Main Chamber Pressure	psi	5.0	4.7	5.9
Ox Tank Pressure	psi	15	15	23
ΔP Oxid. Circuit	psi	8.7	9.8	15.7
Oxid Injector Inlet Pressure	psi	6.3	5.7	7.3
ΔP Oxid. Injector	psi	1.3	1.0	1.4
Total Oxid Flow	lb/sec	.094	.097	.126
Total Fuel Flow	lb/sec	.050	.037	.041
Cooling Jacket Flow	lb/sec	.022	.037	.041
GG and Turbine B.P. Flow		.028	0	0
Preburner Bypass Flow	lb/sec	.015	.026	.028
Preburner Fuel Flow	lb/sec	.007	.011	.013
Mean Temperature CJKT (°R)		520	990	990
Mean Temperature Main Ox Injector (°R)		520	402	402

TABLE XXIV (cont.)

START AND SHUTDOWN TRANSIENT SEQUENCES SUMMARY



IDLE MODE SEQUENCE OF EVENTS

A. START IDLE MODE SEQUENCE OF EVENTS

Pre Childown Idle Mode

1. Preburner bypass valve open
2. Cooling jacket bypass valve open
3. Helium pressurization valve open
4. Oxidizer manifold purge on
5. Interpropellant seal purge on

Propellant Settling and Thrust Chamber Cooling Jacket Chill Phase

1. Fuel prevalve open
2. Oxidizer prevalve open
3. Thrust chamber igniter spark plug energized
4. Thrust chamber igniter valve open
5. Fuel discharge valve open
6. Thrust chamber igniter MR closed loop control on

   Upon ignition of the igniters the igniter closed loop control maintains an igniter MR of approximately 0.3 by sensing igniter wall temperature.

7. Thrust chamber igniter spark plug de-energized

   The hydrogen flow is ultimately expanded through the engine nozzle to provide a propellant settling impulse. Fuel flow only is maintained until the propellant have been sufficiently settled.

Chilldown Idle Mode Phase

1. Oxidizer manifold purge off
2. Oxidizer discharge valve open

3. **Thrust chamber igniter pulsed**

The igniter oxidizer valve is momentarily opened to increase the mixture ratio from 0.3 to 1.0 which produces a hot enough gas stream to ignite the main chamber flow.

4. **Oxidizer discharge valve closed to preset MR position**

The oxidizer discharge valve is positioned to constant thrust chamber MR. This chilldown idle mode is continued until the fuel and oxidizer pumps are sufficiently chilled to provide liquid propellant at the pump inlets.

Idle Mode to Full Thrust

1. Energize preburner igniter spark plug
2. Preburner igniter oxidizer valve opened
3. Preburner igniter ignited
4. Preburner bypass valve programed closed
5. Cooling jacket bypass valve programed closed
6. Preburner oxidizer valve opened
7. Oxidizer discharge valve fully opened
8. Preburner igniter de-energized
9. Preburner igniter oxidizer valve closed
10. Thrust chamber igniter oxidizer valve closed
11. T.C. Igniter Closed loop control system off
12. Helium pressurization valve closed
13. Closed loop control system on

The closed loop control system is off during the thrust rise from idle mode to full thrust. Valve sequencing is time

controlled. Upon reaching 90% rated thrust the closed loop system is re-energized to control thrust and mixture as required during flight.

B. SHUTDOWN IDLE MODE SEQUENCE OF EVENTS

At Shutdown Idle Mode Signal

1. Closed loop control system off
2. Preburner oxidizer valve closed
3. Helium pressurization valve open
4. Preburner bypass valve opened to idle mode position
5. Cooling jacket bypass valve opened to idle mode position
6. Oxidizer discharge valve closed to idle mode position

   The oxidizer discharge valve and bypass valves are positioned to operate the engine at an idle mode thrust and MR.

Shutdown

1. Oxidizer discharge valve closed
2. Oxidizer system purge on
3. Fuel discharge valve closed
4. Helium pressurization valve closed
5. Prevalves closed.

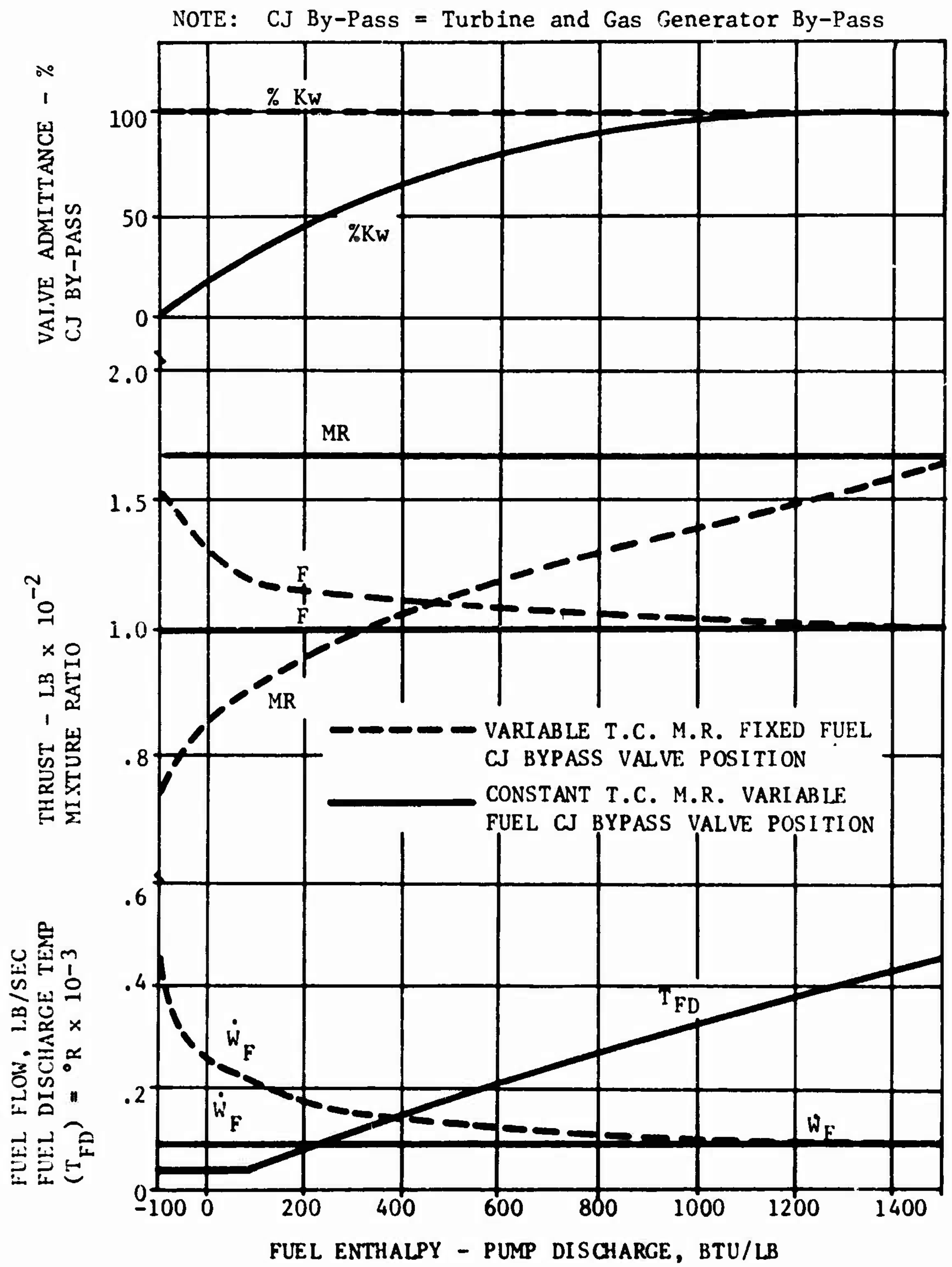


Figure 139. Fuel Pump Childown Parameters

closing the preburner and cooling jacket fuel bypass valves, and sequencing open in the oxidizer thrust chamber valve.

For post shutdown idle mode, the engine is shut down by closing the preburner oxidizer valve and energizing the idle mode control system to open the preburner bypass valve. The fuel and oxidizer discharge valves remain open so thrust decays to an idle mode thrust rather than to zero. The idle mode is terminated by closing the oxidizer discharge valve followed by closing the fuel discharge valve.

The entire idle mode sequence of events is summarized in Table XXI.

g. Engine System Control Methods

The OOS Engine is designed to operate over a thrust range of 25,000 lb to 5,000 lb and a mixture ratio range of 6.5 to 5.5. This range of operation is achieved by modulating the two LO_2 throttle valves and the preburner hydrogen bypass valve shown schematically in Figure 140. The fourth modulating valve, the cooling jacket bypass valve, is used for engine childown and during the start to keep the hydrogen pump out of stall.

There are four possible methods for controlling thrust and mixture ratio shown in the matrix below:

Thrust	Mixture Ratio Open Loop	Mixture Ratio Closed Loop
Open Loop	X	X
Closed Loop	X	X

Open loop control is achieved by commanding the valves to a pre-programmed position, whereas closed loop control is established by continuously modulating the valves to maintain engine performance at the desired operating point. The closed loop control is the most desirable system.

Thrust and mixture ratio are integrally related. If the open loop control scheme is utilized, commanded thrust changes will result in mixture ratio changes. The sensitivity of mixture ratio to change in engine thrust level for modulation of the two steady state control valves is shown below:

Valve	$\frac{\Delta MR/MR.}{\Delta F/F.}$
Ox Preburner Valve	0.272
Ox TCA Valve	-0.230
Preburner Fuel Bypass Valve	fixed

Where: MR = 6.0
F = 25,000 lb

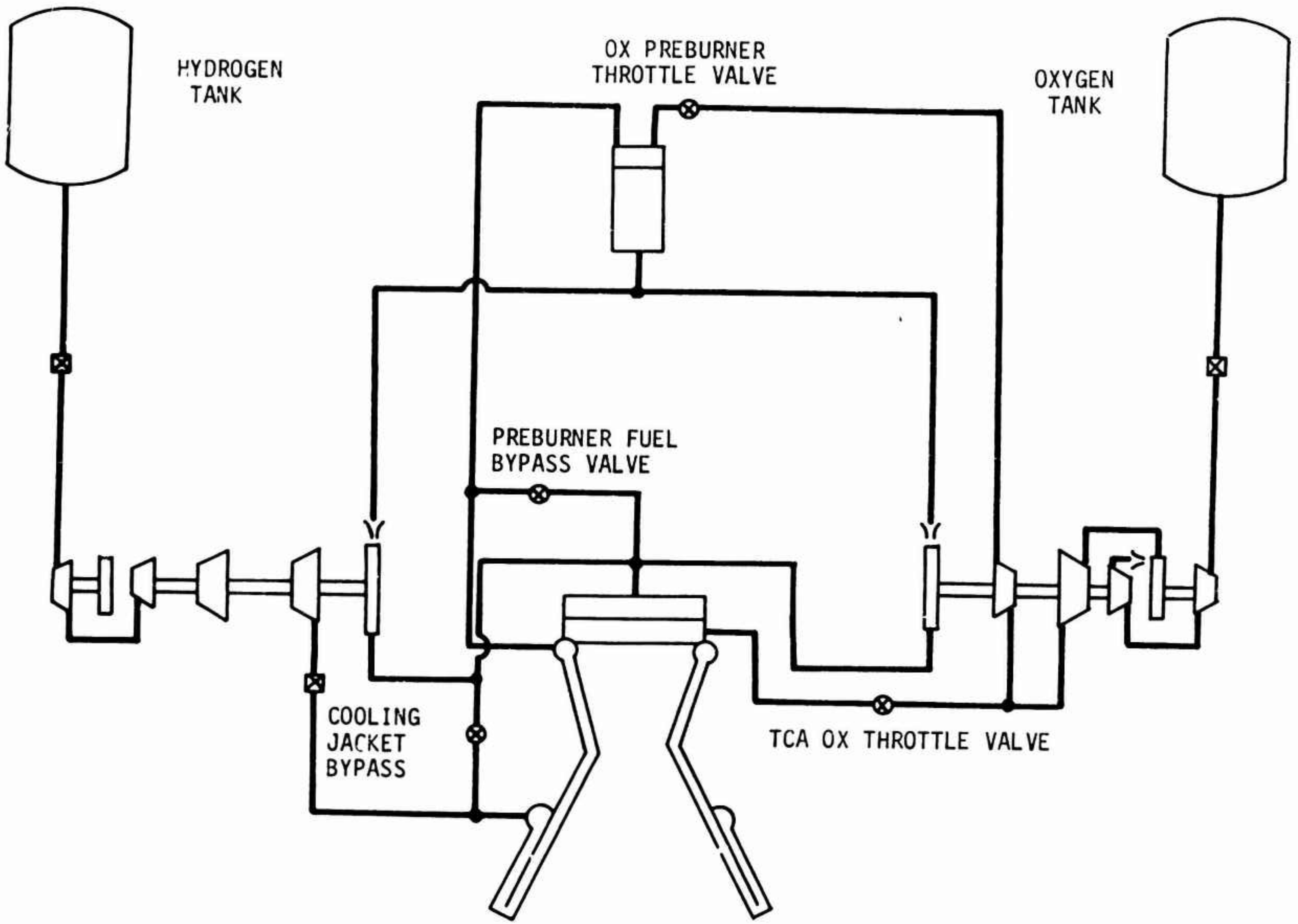


Figure 140. OOS Engine Steady-State Control Schematic

If a ten percent increase in thrust level is desired, it can be obtained by decreasing the preburner valve ΔP by 11% and increasing the Ox TCA valve ΔP by 27%. The resultant valve settings cause the mixture ratio to increase by 0.21%. If changes such as this are acceptable with respect to vehicle propellant utilization, it is recommended that the engine thrust be controlled closed loop, allowing mixture ratio control to run open loop. The controller would then command the valves to control thrust while ignoring the resultant small drifts in mixture ratio. On the other hand, commanded mixture ratio changes would be processed such that valve modulation would send the engine to the desired mixture ratio while maintaining tight control on thrust.

Three conceptual layouts for steady state thrust and MR control (Figures 141, 142 and 143) are presented herein. The first two concepts are control loops contained completely within the engine side of the system interface. The third concept utilizes vehicle data to control the engine, and does not require any engine parameters feedback. The desirable and undesirable features of these configurations are summarized in Table XXV.

The first system utilizes flow meters to measure propellant weight flows. Note that the flow meters are located downstream of the pumps in order to avoid meter-generated cavitation during start. Weight flows would be processed by the computer to calculate engine mixture ratio. Chamber pressure would provide the signal for calculating engine thrust. The error signals generated by the difference between calculated and commanded engine performance would be operated upon by the controller, and the primary combustor ox valve and the secondary combustor ox valve would be actuated simultaneously to correct deficiency in commanded engine performance.

The second system utilizes the boost pumps as flow meters. Suction pressure and temperature, boost pump speed, and discharge pressure are sensed. These data are processed to calculate the head rise across the boost pump. The computer would contain the pump operating map such that pump speed and head rise would be sufficient to calculate weight flow. From this point the system would perform identically to the system first described.

The third system utilizes vehicle sensed data only to control engine performance. Tank level sensors would be used to indicate the propellant flow rates being consumed by the engine. Since the level sensors are inaccurate until tank ullage reaches about 20%, the control valves would be commanded to precalibrated position such that the engine would operate at nominal mixture ratio until sufficient propellant had been consumed. At that point, weight flows calculated from tank level sensors would be processed, and the mixture ratio control would run closed loop for the duration of the mission. An on-board accelerometer would provide thrust information to the computer. The thrust control could run closed loop with measured thrust from the accelerometer, or the system could run open loop compensating for thrust variations by allowing the engine to burn until the accelerometer sensed that the desired terminal velocity had been reached.

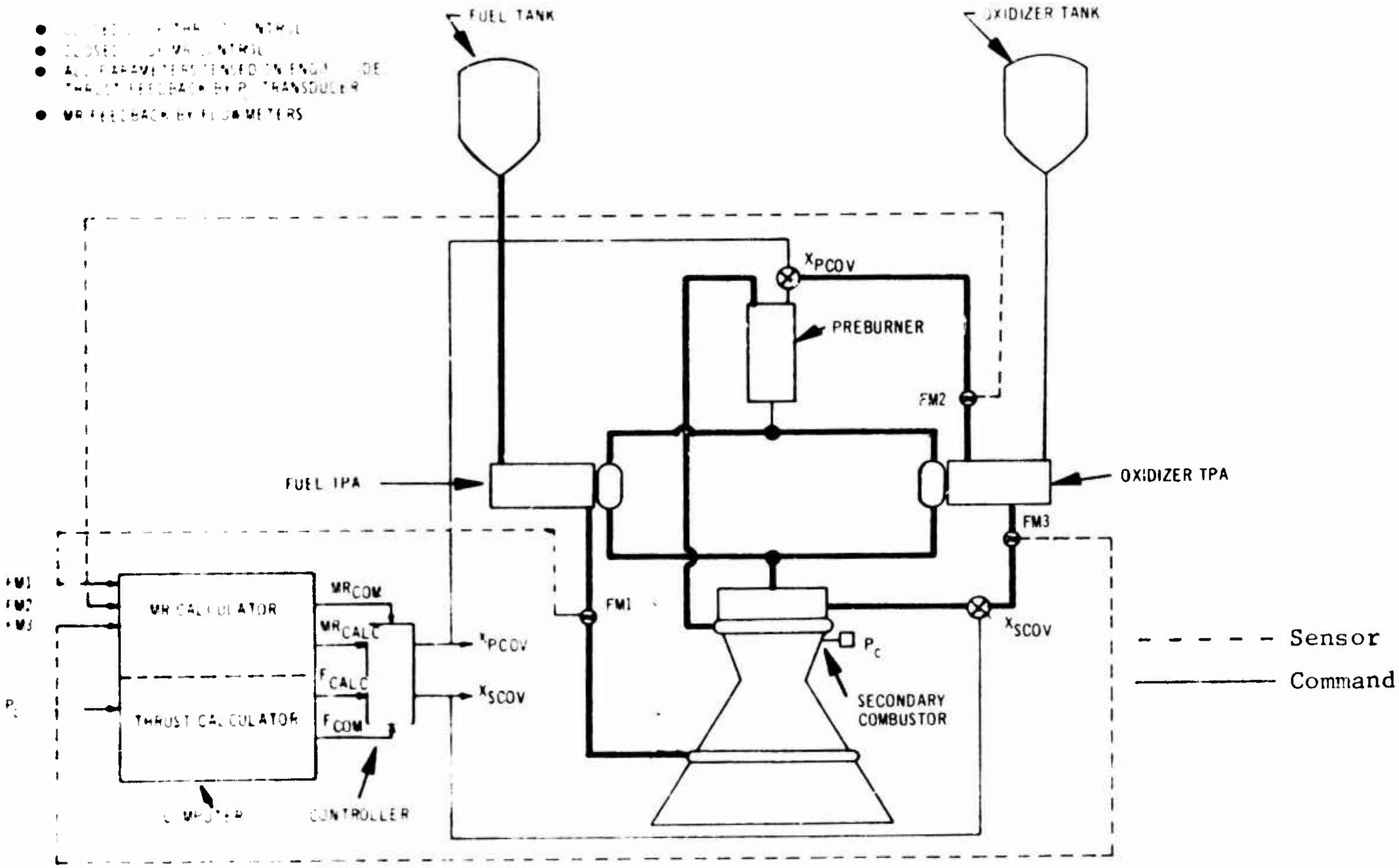


Figure 141. Control System for Steady-State Thrust and Mixture Ratio Control, Concept No. 1

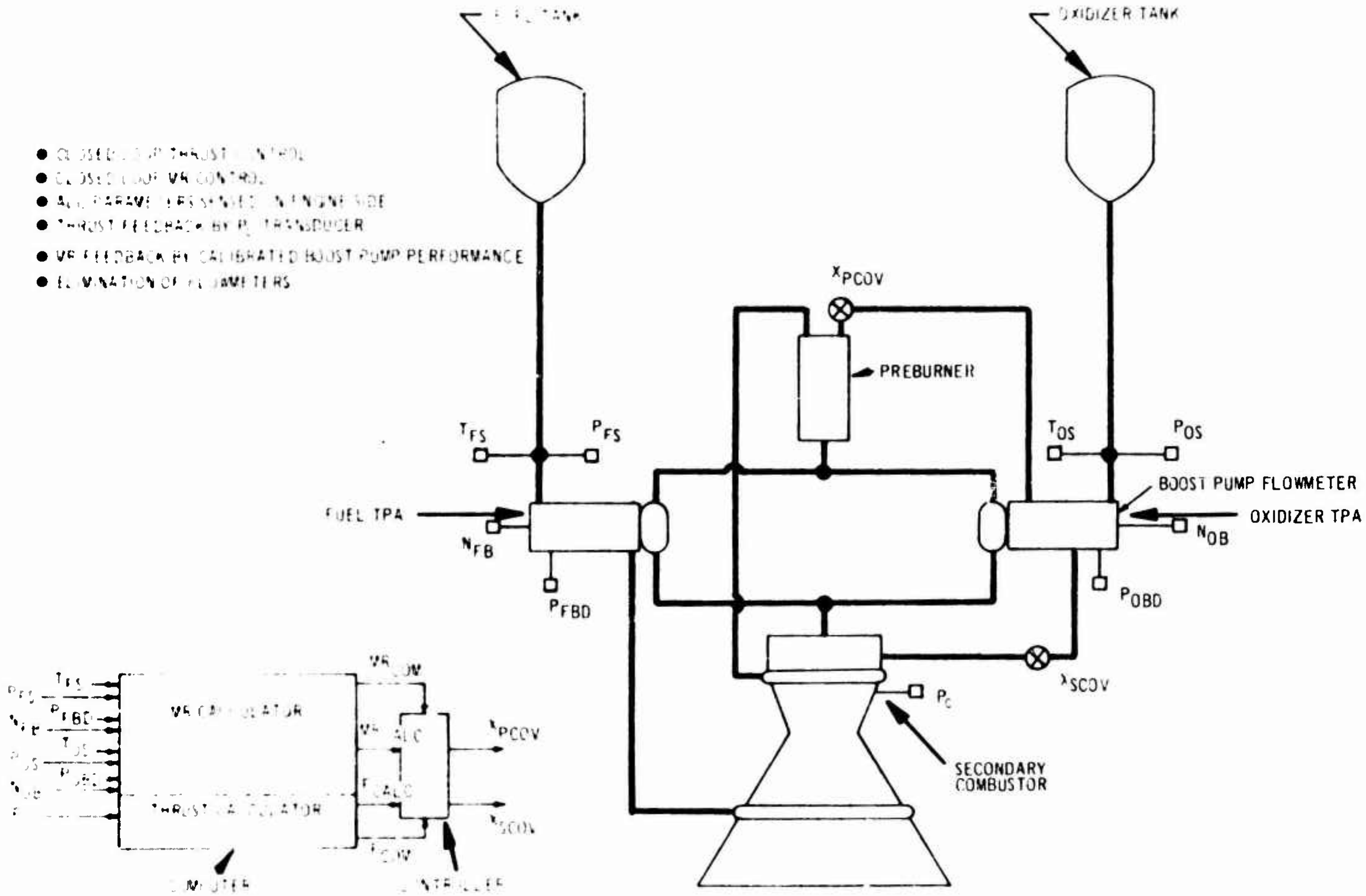


Figure 142. Control System for Steady-State Thrust and Mixture Ratio Control Concept No. 2

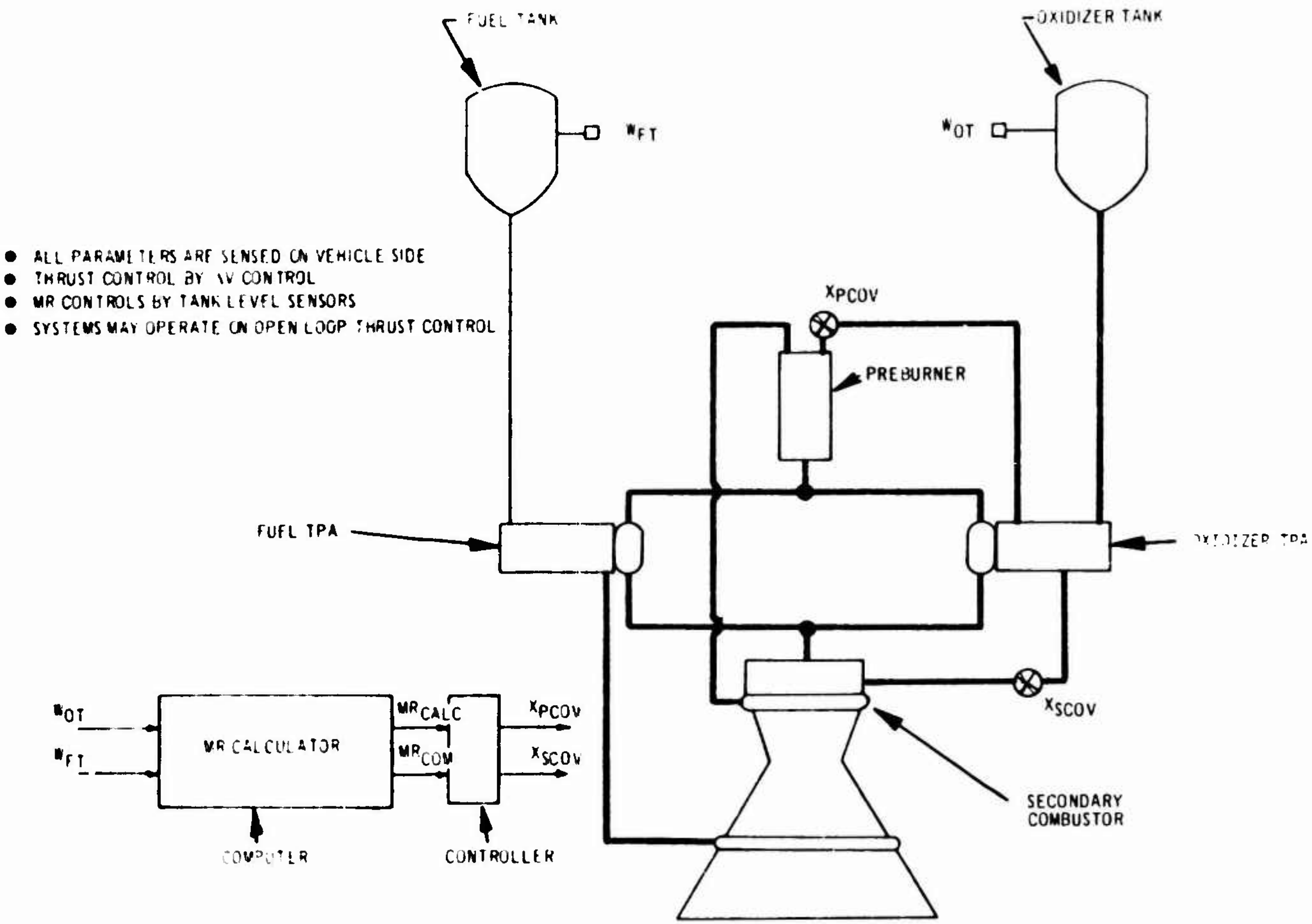


Figure 143. Control System for Steady-State Thrust and Mixture Ratio Control, Concept No. 3

TABLE XXV

STEADY STATE THRUST AND MR CONTROL SYSTEM DESIGN

	DESCRIPTION	GOOD FEATURES	BAD FEATURES
1	Closed Loop Thrust Control, Closed Loop MR Control, All parameters sensed on engine side of interface Propellant flow measured by flow meters	Direct flow rate measurement MR calculation is simple using direct flow rate measurement Chamber pressure transducer would already be part of engine operating integrity instrumentation.	Flow meters add engine weight Flow meters require AC-to-DC signal processing. Flow meters might overspeed during start.
2	Closed Loop Thrust Control, Closed Loop MR Control, All parameters sensed on engine side of interface. Flow calculated from low speed pump characteristic	Lack of flow meters saves engine weight Suction pressure and temperature sensors would already be part of engine instrumentation used to establish engine start sequence. Chamber pressure transducer would already be part of engine operating integrity instrumentation.	MR calculation requires processing of many signals. Computation of MR is complex. Boost pump speeds and discharge pressures would ordinarily not be sensed.
3	Open or Closed Loop Thrust Control, Closed Loop MR Control, All parameters sensed on vehicle side of interface. Flow measured by tank level sensors	Instruments already available on vehicle. Lack of instrumentation on engine saves weight.	MR control might be important early in mission. Measurements might be less accurate. Control system interface design must be negotiated mutually between two contractors.

h. Optimum Area Ratio for Fixed and Retractable Nozzles

Nozzle optimization for the OOS propulsion system is of the utmost importance since the payload capability is sensitive to engine weight. The large area ratio considered for both the fixed and retractable nozzle concept constitute the most heavy single engine component. In the nozzle optimization study, the first step consisted of optimizing the nozzle contour. This was accomplished by evaluating nozzle weight and performance for various bell nozzle contours, Figure 144. The result indicated that the minimum length Rao contour results in the highest payload capability within a fixed engine length.

The second step of the optimization consisted of establishing the optimum area ratio for the given mission and payload sensitivity parameters. For this purpose, engine weight and engine performance relationship was established. Figure 145 and the payload equations applied, which is indicated by lines of constant payload ΔPL lines in Figures 145 and 146.

This nozzle optimization was conducted for the retractable nozzles for both the orbit-to-orbit mission and the lunar lander mission and is only valid for the given sensitivity factors.

The results indicate that the optimum area ratio at mixture ratio 6 and the selected chamber pressure are larger than $\epsilon = 500$ for both missions. The orbit-to-orbit mission optimum area ratios are lower than for the lunar lander.

The results also indicate that it is not practical to design the nozzle exactly at the optimum area ratio since the optimum is very flat. A more practical solution is to sacrifice a very small amount of payload to obtain shorter engines and thus decrease engine gimbal power requirements and actuator weights.

Based on the presented results, it was assumed that a practical nozzle area ratio for the retractable nozzle concept is $\epsilon = 450$ for both missions.

For the optimization, the engine configuration consisted of a regenerative 0.015 tube design from area ratio $\epsilon = 6$ to $\epsilon = 125$ and a radiation cooled nozzle of 1/8-in. AGCarb material.

The nozzle optimization was repeated for design mixture ratio MR = 7.0 and MR = 5.0 presented in Figure 147. The result shows that the optimum area ratio is very insensitive to design mixture ratio.

The conducted optimization studies are sensitive to the nozzle concept and nozzle material used.

The optimization was also made for all regenerative cooled nozzles of various tube thickness and is shown in Figure 148 indicating a considerable impact in payload capability at the larger area ratios.

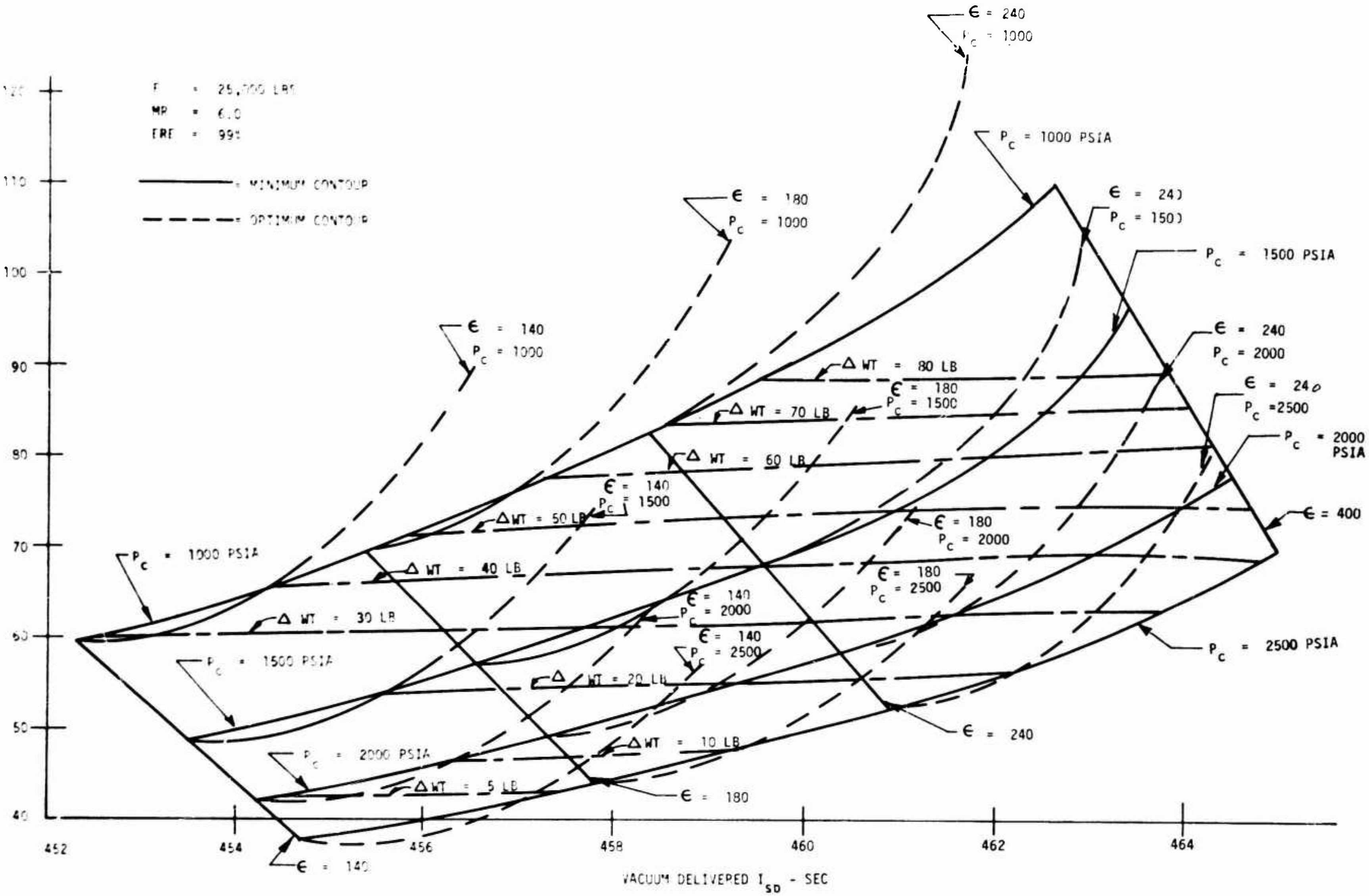


Figure 144. Nozzle Length and Weight Change vs Delivered Vacuum I_s

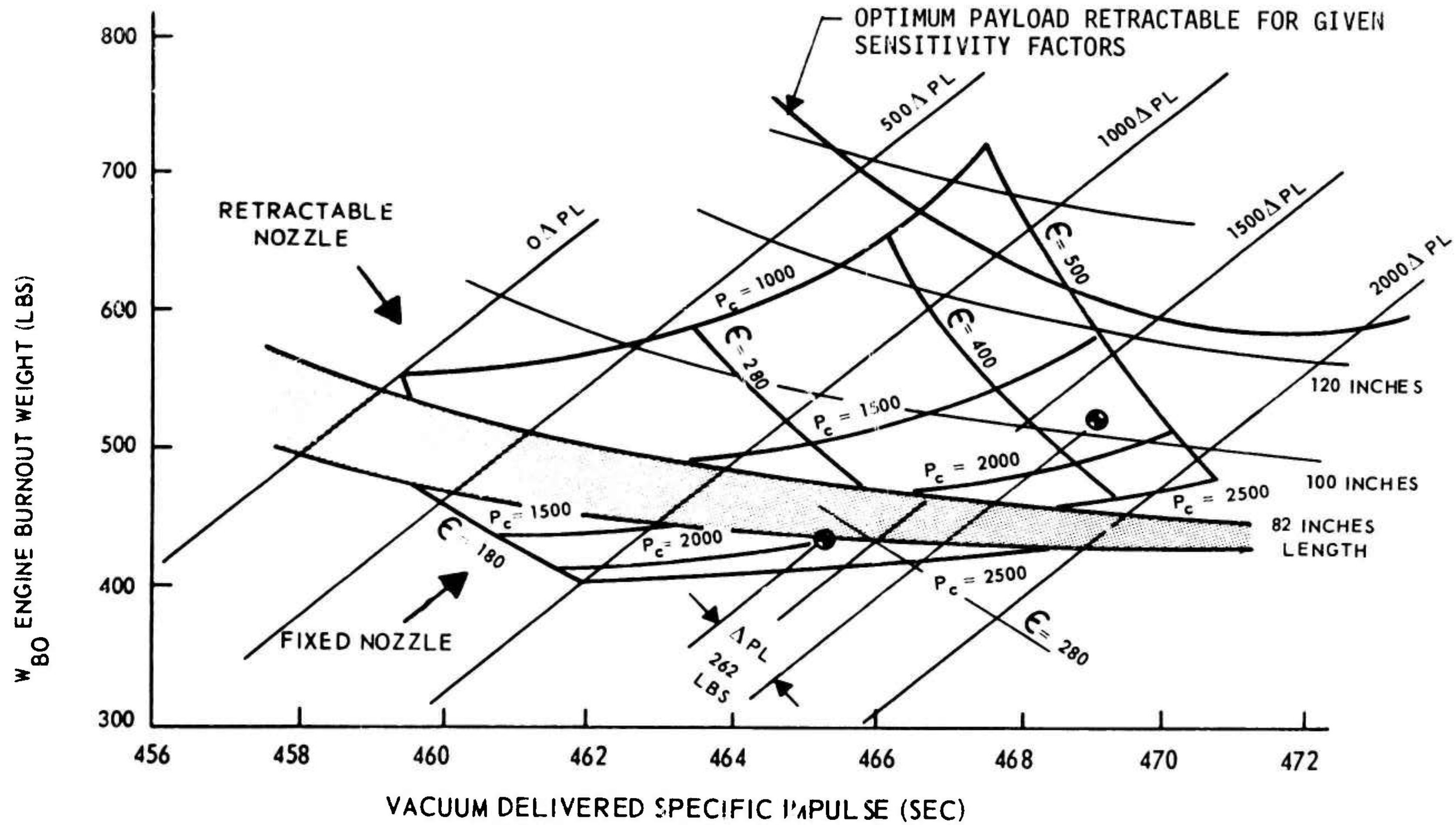


Figure 145. Engine Burnout Weight vs Vacuum Delivered I_s, Orbit-to Orbit Mission

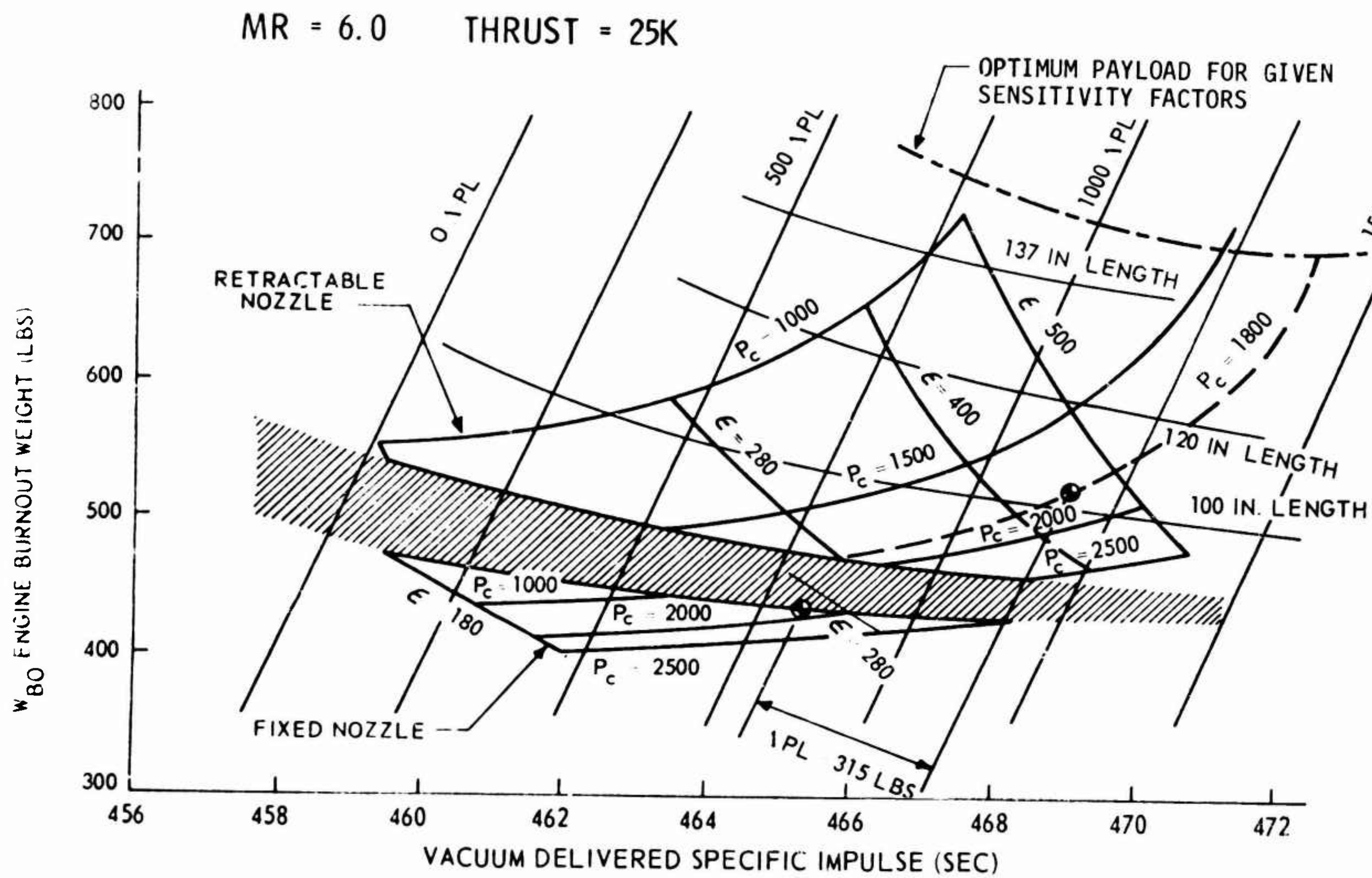


Figure 146. Engine Burnout Weight vs Vacuum Delivered I_s, Lunar Lander Mission

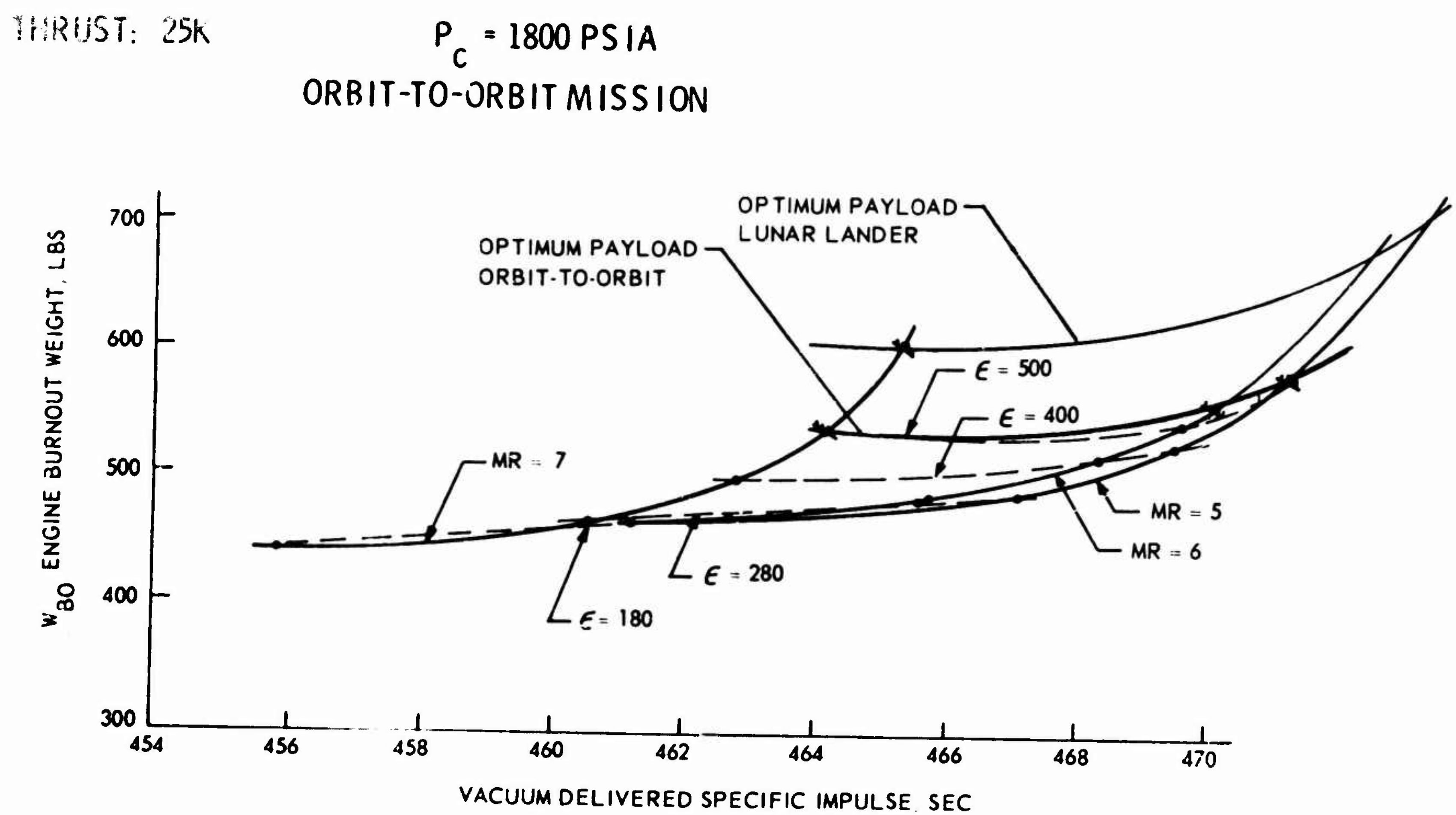


Figure 147. Engine Burnout Weight vs Vacuum Delivered I_s, Minimum Weight Retractable Nozzle

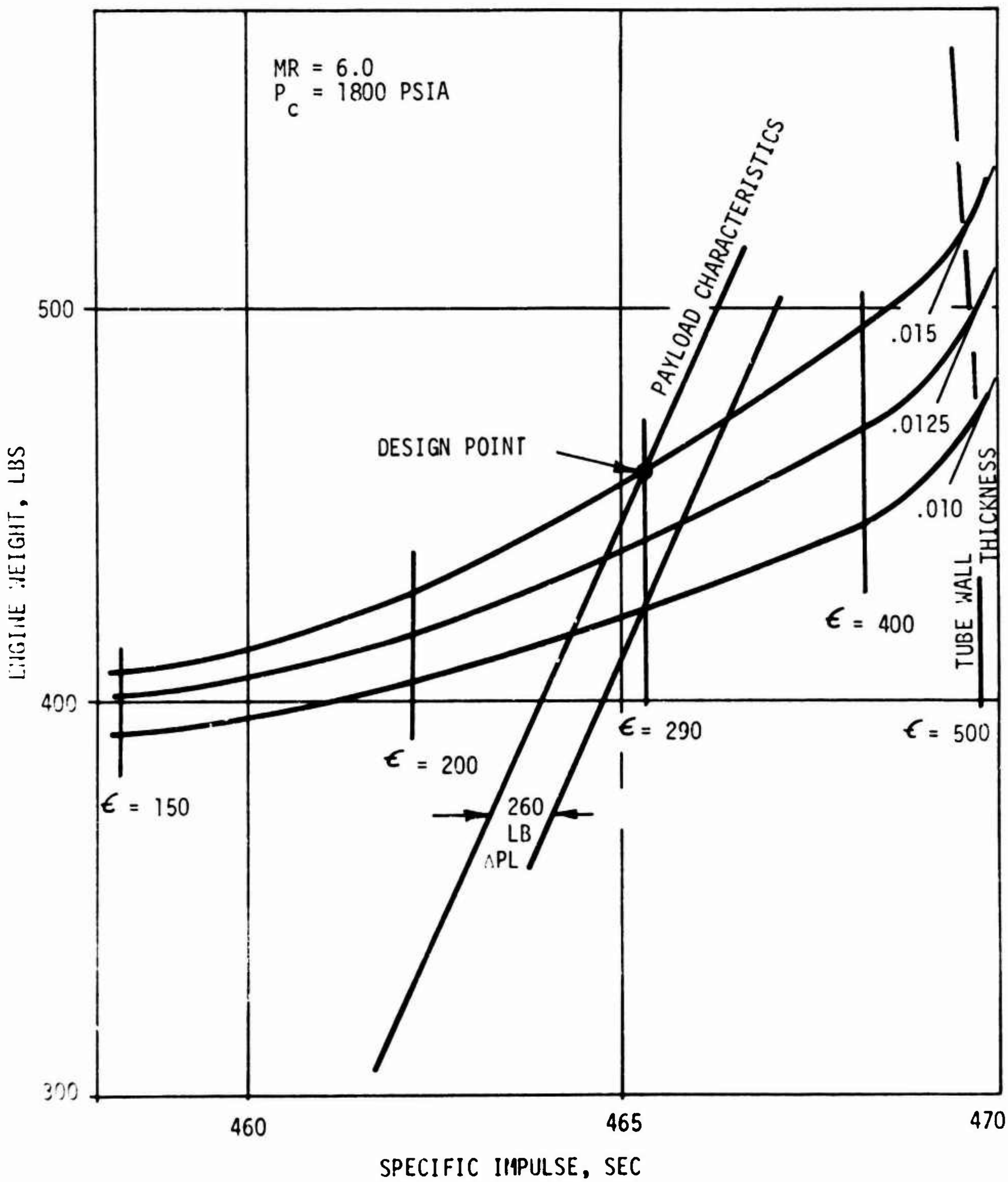


Figure 148. 25K Engine Effect of Nozzle Tube Thickness vs Engine Weight and Performance

For the smaller area ratios for fixed nozzles, the payload impact of tube thickness is relatively small.

For the fixed nozzle concept, the nozzle area ratio optimization is constraint for the OOS application to a fixed engine length of 82 in. The maximum obtainable nozzle area ratio is therefore depending on the required combustion chamber length L'. The relationship of L' and achievable area ratios is shown in Figure 149. The required combustion chamber length is related to the energy release efficiency. It is feasible that increase in chamber length may increase the ERE and actual result in a net performance gain. The actual chamber length L' will be determined experimentally and the presented assumption is based on experience.

The area ratio optimization is somewhat problematic due to uncertainty of the actual achievable performance level of large area ratio nozzles. No large area ratio nozzles have been hot tested to date to correlate performance prediction methods with test data. Also, the optimum tube wall thickness of all regenerative tubes is quite uncertain since practical factors such as handling capability in a reusable system and fabrication cost will force a compromise.

For the baseline engine a tube wall thickness of the all regenerative cooled nozzle of 0.015 in. was assumed. For comparison purposes for the 10K engine, the tube wall thickness of 0.010 in. was assumed since the fixed nozzles for this engine are much more sensitive to nozzle weight.

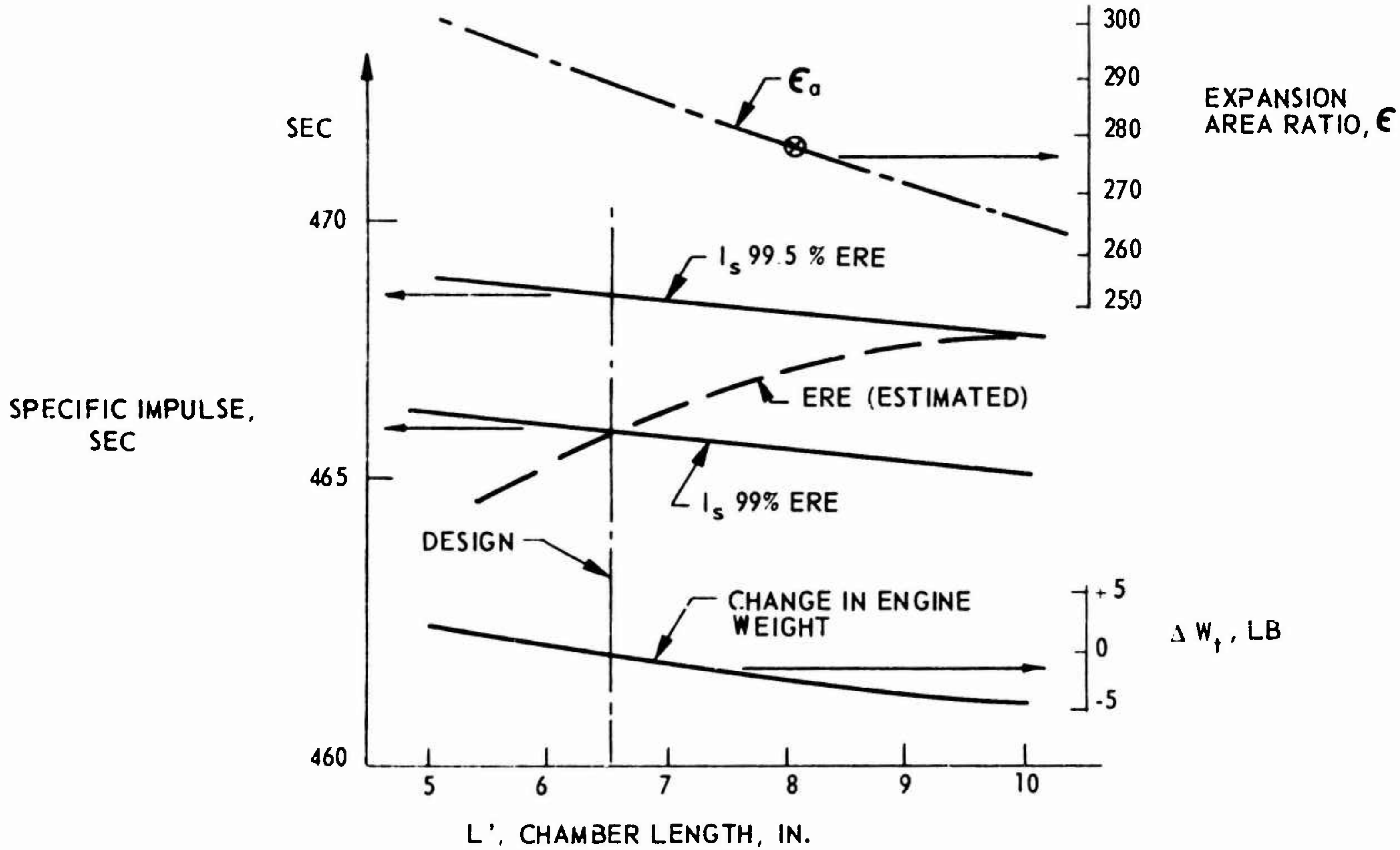


Figure 149. Chamber Length vs Specific Impulse

2. Major Component Design and Description

This section describes the concept details and the results of the engineering design analysis which led to the final OOS engine configuration.

This section also includes the component detail drawings used for the engine weight evaluation.

The 25K engine is composed of the following eight major components:

- Main Injector and Hot Gas Inlet Manifold
- Thrust Chamber
- Nozzle
- Preburner
- Igniters
- Control Valves
- Turbopumps and Low Speed Inducers
- Turbine Hot Gas Manifold and Propellant Lines

The first five components listed above are combustion components, and they, together, form the thrust chamber assembly (TCA). Addition of the last three component groups complete the engine assembly. Figure 150 is a TCA drawing provided to display the relationship between the five combustion component groups in greater clarity than is possible on the engine drawings. More detailed component drawings are presented as well, being referenced in the following component discussions. Each discussion contains functional and physical descriptions of the hardware, its analytically predicted operational characteristics, and justification of the selected concept. Individual analyses are presented in the report appendices located in Section V. Combustion component weights are given in Table XXVI.

a. Main Injector and Hot Gas Inlet Manifold

The primary function of the main injector is to deliver a mixture of fuel-rich hot gas (FRHG) and oxygen to the forward end of the thrust chamber in such a way that it may be ignited and efficiently combusted in a minimum length combustor, which has minimum cooling requirements. The combustion must proceed in a smooth, stable, repeatable manner without damaging the thrust chamber or injector itself, over a period of operation as long as 50 hours, including 1500 thermal cycles. The injector is required to be an effective heat exchanger, transferring heat from the FRHG to the oxygen to allow good throttling performance. A third function of the main injector is to serve as a structural member; acting as a support for the oxidizer delivery manifolding, locally containing the thrust chamber pressurized propellants, and as a mechanical connector between the thrust chamber and the FRHG inlet manifold.

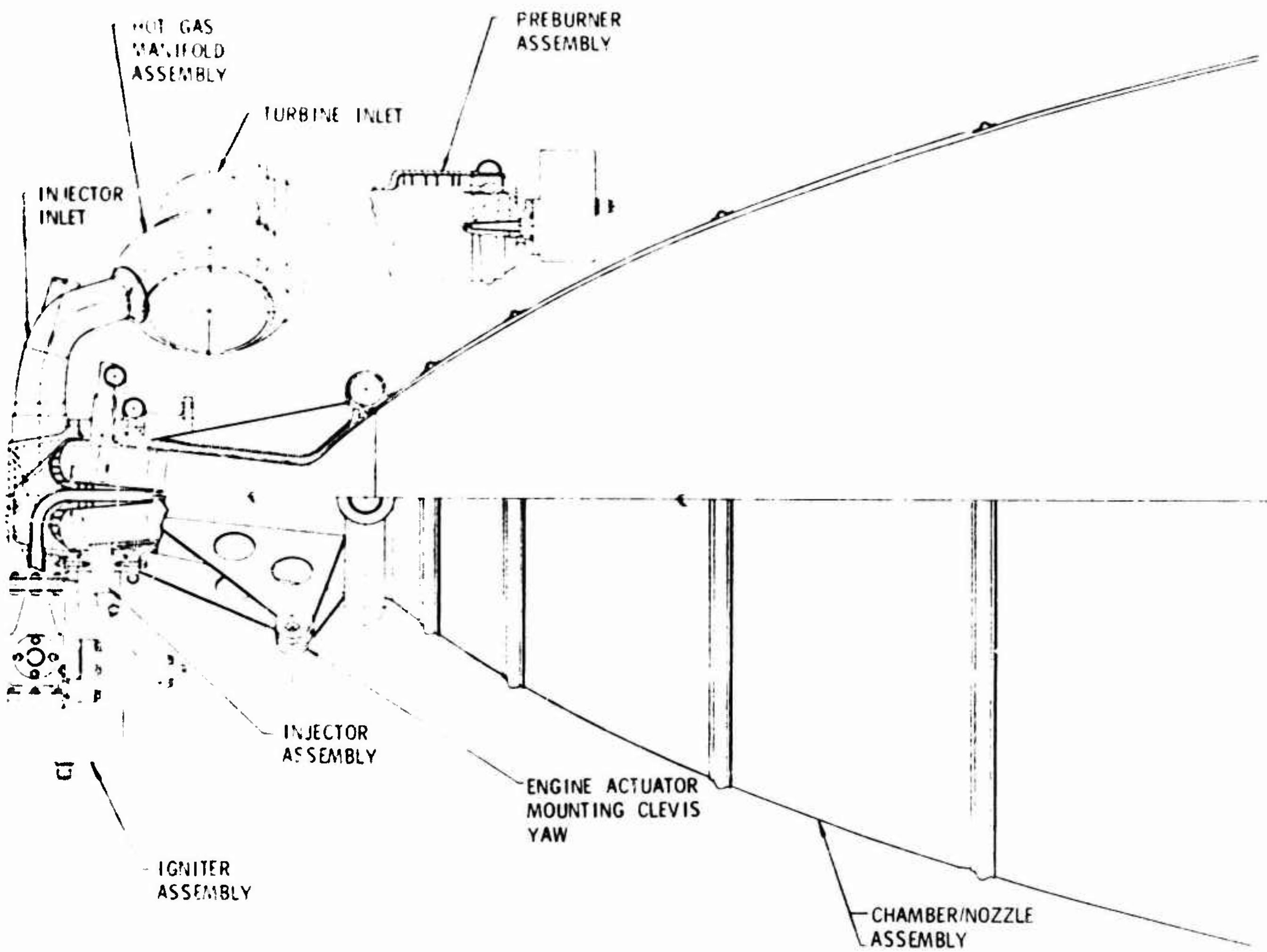


Figure 150. OOS Thrust Chamber Assembly

TABLE XXVI

OOS COMBUSTION COMPONENTS WEIGHT SUMMARY

	Weight, lb	Cumulative Weight, lb
Combustion Chamber Assembly (Cu Liner, Outer Shell, Wire-Wrapped Jacket, Coolant Torii and Flange)	37.1	37.1
Regenerative Tube Bundle	107.4	144.5
Main Injector and Oxidizer Torus	21.7	166.2
Preburner Assembly with Igniter Flange	18.2	184.4
Main Igniter	4.6	189.0
Two Each Dual Exciters	13.0	202.0

Main Injector Description

The main injector is composed of a variable flow area, toroidal oxygen manifold, sixty-four injector vanes, and an injector body. The body is a ring of rectangular cross section, slotted to receive the injector vanes and threaded for bolt-attachment to the thrust chamber and the FRHG inlet manifold. The injector vanes are photo-etched and brazed, "L" shaped flat plate laminates, which receive cryogenic oxygen at their outer extremities, and deliver heated oxygen in the form of like-on-like fans at the plane of the injector face. The vanes form a partial blockage in the FRHG duct formed by the thrust chamber and injector body, such that the slots between and formed by the vanes provide "showerhead" FRHG injection passages. An annular slot formed by the vane outer edges and the thrust chamber wall effects a FRHG barrier cooling circuit for chamber wall auxiliary temperature control.

The vaned configuration of the injector is expected to enhance transverse mode combustion stability in the thrust chamber. A series of acoustic tests conducted on the ALRC SSME injector/chamber indicated that the presence of the vanes caused an order of magnitude increase in transverse mode damping. It was also very difficult to identify any transverse acoustic modes, which gives a good indication of the effectiveness of the vanes as baffles.

To insure that dynamic transverse mode high frequency combustion stability is obtained in the OOS thrust chamber, the eight radial injector vanes extend 0.5 in. beyond the injector face to form injector baffles. These baffles inject oxidizer at their tips, as do the other fifty six vanes. The resultant combustion chamber length to the throat plane from the active baffle tips is 6.0 inches, the minimum length calculated to allow 99% energy release efficiency in the thrust chamber. The required baffle length is estimated to be 0.5 inches, because of the high performance characteristics of the injector. The baffles are required to shield only the pressure/velocity sensitive portion of the combustion process in order to be effective. For a gas/gas injector the sensitive combustion preparation process consists only of primary injectant mixing; this occurs very close to the injector face because of the element type and size. Vaporization sensitivity to local chamber conditions is not a factor, as it would be with gas/liquid or liquid/liquid injection schemes.

The main injector design is shown in Figure 151, and its basic design specifications given in Table XXVII. The design features 64 oxidizer injection vanes grouped in eight identical segments. They contain a total of 640 impinging doublet and 72 canted showerhead elements. The injector pattern is shown in Figure 152; it shows how the like-on-like doublets are staggered along the trailing edge of the vanes, which locate the injector face. Also shown are the showerhead orifices located at the inner vane tip ends which are employed to cool the tip locally and to control the local injection mixture ratio. Showerhead orifices are not utilized at the outer

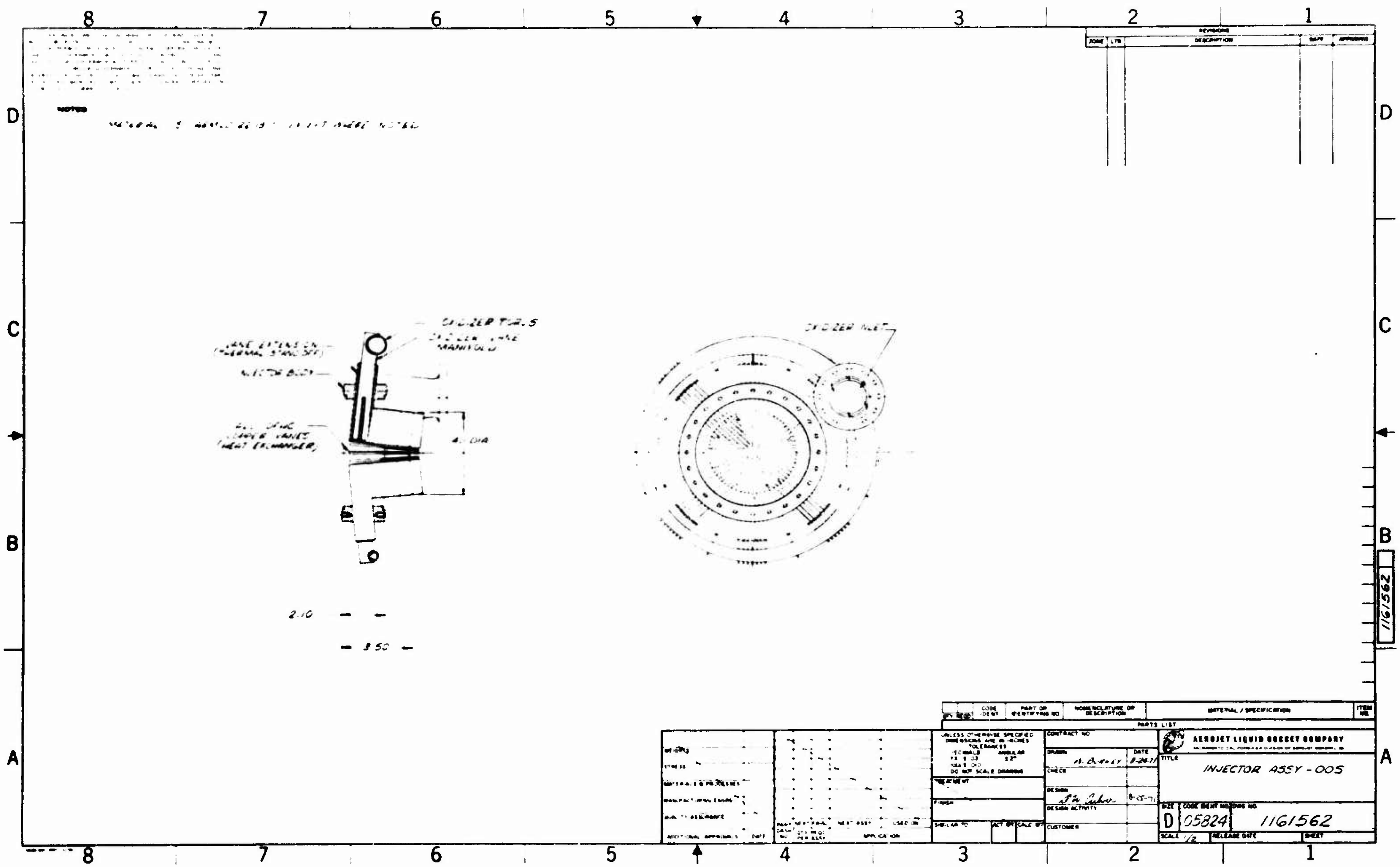


Figure 151. 25K Engine Injector Configuration

TABLE XXVII

MAIN INJECTOR BASIC DESIGN SPECIFICATIONS*

NUMBER OF VANES	64	THRUST PER ELEMENT, LBF	35.1**
NUMBER OF BAFFLES	3	FUEL RICH HOT GAS INJECTION VELOCITY, FT/SEC	575
NUMBER OF ORIFICES	1352	OXIDIZER INJECTION VELOCITY, FT/SEC	415
ORIFICE SHAPE AND SIZE,		FUEL RICH HOT GAS INJECTION TEMPERATURE, °R	1565
RECTANGULAR, IN.	0.0187 x 0.0374	OXIDIZER INJECTION TEMPERATURE, °R	550
TYPE AND NUMBER OF ELEMENTS		INJECTOR WEIGHT, LB	21.5
IMPINGING DOUBLETS	640	AXIAL LENGTH, IN.	3.5
SHOWERHEAD	72	INJECTOR FACE DIAMETER, IN.	4.1
TOTAL ELEMENTS	712	OXIDIZER PRESSURE DROP, PSI	600
DOUBLET IMPINGEMENT ANGLE, DEGREES	60	FUEL RICH HOT GAS PRESSURE DROP, PSI	140
DOUBLET IMPINGEMENT DISTANCE, IN.	0.052		
STAGGERED DOUBLET CENTERLINE SPACING, IN.	0.093		
VANE CENTERLINE SPACING, IN.	0.193		

* DATA FOR 25K LBF THRUST AT 6.0 ENGINE MIXTURE RATIO

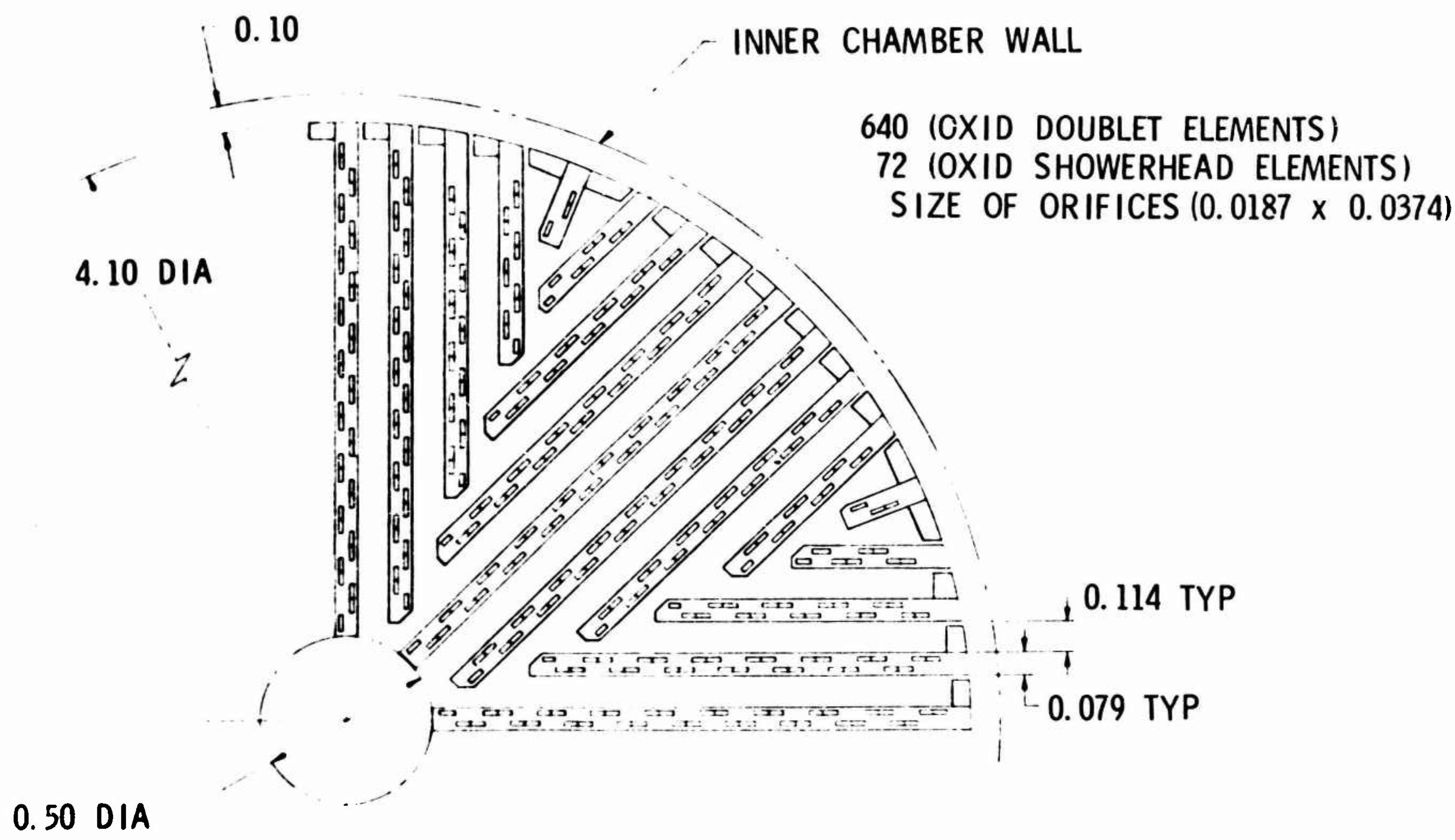


Figure 152. OOS Injector Face

tip ends because of the presence of a barrier coolant injection annulus at the injector priphery. Oxidizer injection at that point is not needed because of the low temperature local mixture ratio of 0.85, and it is not desired because it would contribute to the destruction of the cool chamber boundary layer. Oxidizer doublets are used along the middle portion of the vane tips so that the resulting tangential spray fans force mixing between the two propellants by penetrating the inter-vane sheet of FRHG.

Hot Gas Manifold

The hot gas inlet manifold is composed of an outer, structural conic frustrum, and an inner sheet metal liner. The structural cone is closed at its forward end to contain the FRHG pressure and to form a mount for the engine gimbal block. Its open aft end is flanged for attachment to the main injector body. It is perforated at three side locations to allow introduction of hot or cold fuel gas and to accept a torch type igniter flame tube.

The inner, uncooled, liner forms the FRHG flow passage. Its function is to direct the fuel gas from the dual inlet ports, through a distribution torus and through two gas distribution plates, to the main injector inlet. The high velocity FRHG therefore does not contact the conical structural member, so that its equilibrium temperature remains low during operation, and good structural material properties are maintained. The volume between the inner liner and the outer wall is pressurized by the FRHG through a single annular leak path at the injector-manifold interface, so that gas pressure induced loads are not reacted by the heated liner. The liner also forms the inner wall of the main injector FRHG duct by penetrating the center of the injector to the face plane, thereby also effecting an end support for the igniter torch flame tube. In the region of the FRHG primary distribution torus the liner is double walled. The inner wall is perforated, in order to form a large area, wide band acoustic liner. The purpose of the liner is to damp acoustic perturbations emanating from the turbine hot gas manifold and/or the thrust chamber, and thereby enhance longitudinal mode combustion stability in the thrust chamber.

The hot gas inlet manifold, has several functions. Physically, it provides a mounting pad for the engine gimbal block and mates with the main injector. It receives FRHG from the fuel and oxidizer turbine exhaust ducts during operation, and from a fuel bypass line during starting. It distributes these gases properly to the injector inlet in a minimum length envelope, with minimum pressure and thermal losses. Provision is also made in the FRHG inlet manifold design for a thrust chamber igniter. The manifold life must be equal to or greater than that of the main injector in order to optimize engine maintenance procedures.

A cross sectional view of the two subject subcomponents is shown in Figure 153, assembled with the thrust chamber igniter, inlet lines, and thrust chamber forward section.

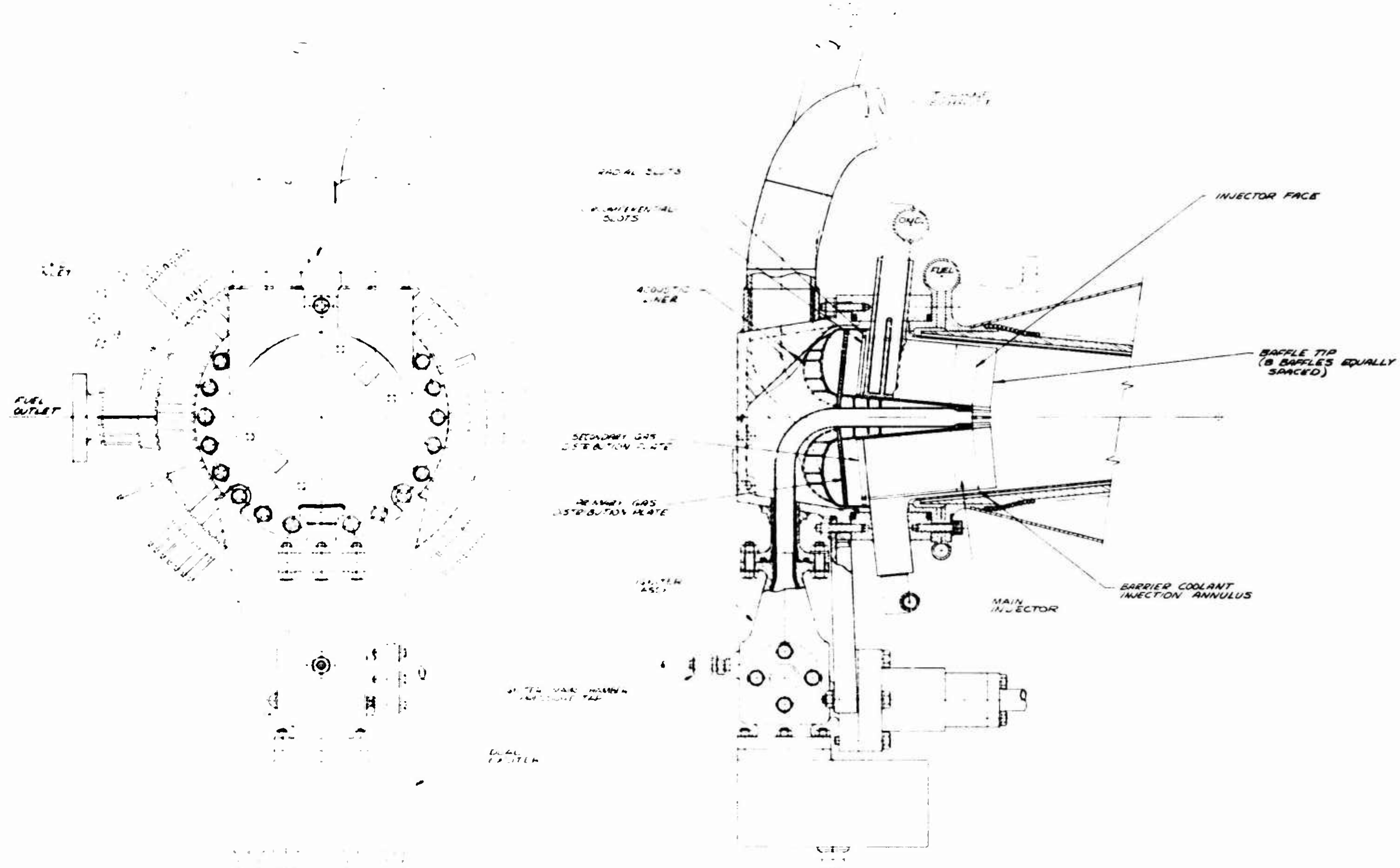


Figure 153. OOS Injector and Igniter Manifolds

Injector Concept Selection

The choice of a main injector design concept is affected by three considerations:

(1) Conditions of fuel and oxidizer at the injector inlet.

(2) Conditions of the oxidizer at the injector outlets.

(3) Type of injection elements to be used.

The design goal is to provide a 5 to 1 throttleable injector which minimizes the following injector and thrust chamber parameters:

(1) Weight

(2) Axial Length

(3) Performances Losses

(4) Cooling Requirements

(5) Maintenance Cost

(6) Fabrication and Development Cost

Given a staged combustion engine cycle, the normally available injector inlet propellant conditions are established as: FRHG at approximately 1700°R, and liquid oxygen at approximately 200°R. The condition of the oxidizer at the injector face plane is that of a supercritical fluid at chamber pressures above 750 psia. This begs the real question however, which is: during throttled operation below 750 psia, what is the state of the injected oxygen; gaseous, two phase, or liquid? It will be shown that two phase injection causes large MRD performance losses. Liquid oxygen injection requires either a complex injector and control valve, or else a high inlet pressure at full thrust to provide adequate injector stiffness at minimum thrust levels. It also requires insulation between the propellants to prevent local oxidizer boiling (heating at supercritical pressures). The liquid injection approach very likely increases the engine maintenance cost requirements as well, because it imposes thermal conditions which do not favor long fatigue life, e.g., high temperature components with large thermal gradients. The proper use of gaseous injection minimizes the above problems. Given, then, a desire for gaseous oxidizer injection, the next question is: how is the oxygen to be vaporized, in the injector, or in an external heat exchanger? An external heat exchanger is <u>not</u> favored for several reasons. First, a large amount of heat is required, and it can be obtained only from heated hydrogen coolant in the chamber or nozzle, or from the FRHG generated in the preburner. Use of the FRHG is favored, for it has the higher temperature, and the required heat exchanger weight and size is therefore, smaller. In addition, the nozzle is physically remote from the injector, so that large high pressure manifolding would be required to transfer gasified oxygen from a nozzle heat exchanger to the injector inlet. It is also doubtful if much heat could be extracted from the nozzle coolant, because it enters at a temperature below the liquid oxygen critical temperature and exits at a temperature only slightly above it. The

chamber is an adjacent component with a backside temperature approximately 200°R above the oxygen temperature and would seem to be a likely heat source for oxygen heating, but the chamber is the single component most prone to low cycle fatigue failure. Any additional back side cooling provided by the liquid oxygen would only increase its wall thermal gradient induced strain. In order to meet minimum life requirements its internal wall temperature would need to be reduced, and this would necessitate greater barrier cooling performance losses.

As the FRHG is the logical heat source, and since it must pass through the injector, the optimum design requires that the injector itself be the oxidizer heat exchanger. In this way no extra liquid or gas oxidizer transfer lines are needed, and no extra components or component interfaces are required. Liquid oxygen is introduced through a liquid control valve directly from the pump outlet to the injector inlet, receives heat from the FRHG and enters the thrust chamber as a heated supercritical fluid or as a gas, but never as a liquid or two phase fluid. This concept has the maximum thermal response during engine start, also, because a minimum mass of metal need change temperature. An all copper injector, therefore, indicated, since it resists injector face erosion and heats the oxygen efficiently, because it is a good thermal conductor.

Injector Element Selection

The remaining design problem is the type of injection element to be used. Two types are worthy of consideration because of the considerable fabrication, analytical, and test experience obtained with them to date. They are the co-axial element, and the impinging element. While much experience is available for the gas/liquid coaxial element, less is available for the gas/gas type. Almost none is available for a FRHG/gas co-axial element, especially where gross heat exchange is required within the injector. Conversely, ALRC has conducted analytical and testing programs recently with impinging gas/gas hydrogen/oxygen injectors as well as with platelet heat exchanger injectors.* A vane type injector is a natural heat exchanger, having two flat sides per vane in contact with high velocity FRHG, and containing virtually any desired internal flow geometry for the oxidizer. The natural injector pattern used with this type of injector is like-on-like oxidizer doublets, oriented in a nominal tangential fan arrangement, intersecting showerhead FRHG sheets formed by flow between the vanes. Performance and injector/chamber compatibility analyses and test data generated at ALRC show this design concept to be very satisfactory for operation at chamber pressures above 2500 psia. Table XXVIII shows test data from ALRC SSME vane subscale testing. Note that both chamber pressure and FRHG mixture ratio be greater than that required by the subject injector.

In contrast to the co-axial element, the impinging pattern does not require high shear velocities in the chamber to induce primary propellant mixing. As a rule of thumb, the chamber length required with a vane injector is approximately one half of that with a coaxial injector of

*SSME APS IR&D

TABLE XXVIII

ALRC SSME THREE VANE SUBSCALE TEST DATA AT HIGH PRESSURE

Vane type:	All-stainless tipped	1 stainless - 2 copper tipped
Test No.:	SP-50-125	SP-50-126
Test Date:	3-24-71	3-25-71
Duration above 90% chamber pressure, sec:	0.35	0.35
Chamber pressure, psia:	2630	2648
Main Injector oxidizer inlet pressure, psia:	3180	3415
Main Injector oxidizer flowrate, lb/sec:	7.08	7.60
Main Injector mixture ratio at active vanes:	5.60	6.00
Primary gas flowrate, lb/sec:	19.07	19.08
Primary gas mixture ratio:	0.98	0.96
Vane tip condition after test:	heavy corrosion	stainless eroded - copper no damage

equivalent ERE. The proposed injector with gas/gas injection can contain a much larger number of elements in a smaller injector face than can the co-axial type. Compared to a coaxial injector, therefore it reduces both the chamber length and diameter requirements, and, therefore, engine weight and cooling requirements as well. Table XXIX compares the FRHG/Gas Coaxial and vane type injectors, and shows the size and weight advantages of the selected design. An additional factor of the coaxial design is that it normally requires an injector face, which must be transpiration cooled. The selected design type has no such requirement.

The number of elements which can be used per vane is not limited by oxidizer spray fan interference, but, rather, by inter-element geometric interference of the orifices at their inlets. By staggering the doublets, it was found that nearly twice as many elements could be utilized without interference, especially since the orifice sizes decrease to 0.044 x 0.022-in. rectangles. If the doublets had not been staggered, it would have been necessary to decrease the impingement angle from 60° to nearly 30° to accommodate the same number of elements. This would narrow the spray fan angle and delay the propellant mixing, possibly to the point of reducing performance regardless of the chamber length. It is not possible to reduce the impingement distance from its design value of 0.052 in. to increase the allowable number of elements.

The doublet elements are staggered to increase the overall engine performance. Weight and performance analyses indicate that the use of chamber contraction ratios greater than approximately 2.0 and, hence, large injector face areas act to increase injector and chamber weights. This is because the weight increase due to the larger component diameters required are not necessarily offset by shorter axial length, or a given level of energy release efficiency. In fact, if the injector thrust per element value is held reasonably constant, the high chamber velocities resulting from a low contraction ratio design provide beneficial secondary mixing effects. Below a contraction ratio of approximately 2.0, however, the total pressure loss caused by heat addition at high Mach number becomes significant and has an undesirable effect on the engine pressure schedule. Optimum design, therefore, calls for a large number of elements effectively located in a small injector face area. The use of a high chamber pressure of 1800 psia, and the existence of both an igniter torch flame tube and a barrier cooling circuit in the injector face causes the active injector face outside diameter to be 4.1 in., and the inside diameter to be 0.5 in. The number of injector vanes which can be used is dictated by pressure schedule, structural, heat transfer, and performance considerations, which will be discussed below.

Doublet staggering has another desirable feature. One of the inherent features of a vane type injector is its small metallic injector face area, and reduced injector face cooling requirements. The subject injector utilizes OFHC copper vanes such that heat transfer is enhanced in the heat exchanger portion of the vane, and the tips may be regeneratively

TABLE XXIX

COMPARISON OF COAXIAL AND VANE INJECTOR FOR OOS ENGINE

	Gas/Gas Co-Ax	Gas-Gas Vaned	Ratio Vaned Co-Ax
Metal Face Area	11.0	7.5	.70
Axial Length, in.	3.0	3.0	1.0
Chamber Contraction Area Ratio	2.5	2.0	.80
Chamber Dia, in.	4.8	4.3	.90
Chamber Area, in.^{2}	18.0	14.5	.80

cooled by the 415 ft/sec oxidizer injection flow velocity. The heat transfer characteristics of the injector require that the vane thickness be as great as 0.156 in. near its forward end. The vane thickness at the vane tip is 0.079 in. This requires that the external vane cross-section be tapered in the tip region. Since FRHG flows between the vanes, the tapered vane assembly forms a subsonic diffuser for the FRHG. In order to preserve total FRHG pressure and to avoid hot gas flow separation from the vanes prior to oxygen injection, the diffusion angle must be as small as 5°. The use of a very narrow vane tip would then, require a lengthy tip section. A major portion of the tip section is not useful as an effective heat exchanger, because of the variable wall geometry in that region. Therefore, the use of a lengthy tip would increase the axial injector length. This would reduce engine performance by decreasing the length and area ratio of the fixed nozzle, which must fit within the prescribed envelope. Staggering the oxidizer doublets allows the use of a wider vane tip than would be possible with inline elements, because it simulates two inline tips joined together, and therefore provides as effective tip cooling as would a tip one half as thick. The use of a thick tip, then, in conjunction with the thicker vane body, allows a short axial length tapered vane tip section design, and reduces the required injector axial length substantially.

A further reduction in the required vane tip section axial length is effected by the overall geometry of the injector. Because the thrust chamber is conical, the selected injector face geometry is that of a large central angle conic surface so that the vane tips are perpendicular to the chamber wall. The injector vane assembly, therefore, forms a perforated conic frustrum with conical ends; it is perforated by a conical liner surrounding the torch type igniter flame tube. Since the injector face area is smaller than the injector inlet area, some vane tapering can be accomplished without diffusing the FRHG. Therefore, the actual tip taper angle is greater than the gas diffusion angle, and the tip section is shorter than would a corresponding tip on a similar non-conical injector.

Injector Operational Analysis

Table XXX lists the main injector operating conditions as functions of engine mixture ratio and percent of maximum thrust. The detailed data was generated to illustrate how the heat exchanger feature of the injector alters the oxidizer density and allows the retention of favorable injection characteristics over the 5 to 1 throttling range and the required off design mixture ratio range from 5.5 to 6.5 to 1. In order to demonstrate the effectiveness of the concept it was assumed that the preburner fuel turbine bypass valve was not used, e.g., that it remained closed

Table XXX. Main Injector Operating Conditions vs MR and Thrust

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	ρ (lbm/ft³)	Δ (psia)	Δ [illegible] %	[illegible] (psia)	[illegible]	ρ_{FRHG} (lbm/ft³)	ΔP_{FRHG} (psia)	$\Delta P_{FRHG}/P_c$ %	[illegible] (psia)	V_{FJ} (ft/sec)	V_{OJ} (ft/sec)	V_{FJ}/V_{OJ}	$\bar{M}_{FRHG}$	K_{FRHG}
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.360	140	0.075	1976	645	416	1.55	3.67	1.36
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	400	0.271	1587	1585	0.300	[illegible]	0.064	1581	584	379	1.54	3.43	1.37
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4.1	[illegible]	[illegible]	1344	1395	0.234	60	0.054	1170	525	315	1.67	3.16	1.38
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	137	[illegible]	677	1210	0.166	32	0.043	772	453	248	1.83	2.91	1.38
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	675	1105	0.129	21	0.035	571	418	286	1.46	2.76	1.39
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.151	426	1005	0.069	12+	0.033	382	386	200	1.93	2.60	1.39
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.274	2363	1730	0.366	158	0.085	2213	680	405	1.68	3.66	1.36
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.258	1968	1551	0.305	107	0.072	1592	611	370	1.66	3.42	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4.1	[illegible]	0.205	1337	1355	0.241	63	0.057	1173	530	310	1.71	3.16	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7.4	141	0.177	872	1165	0.172	35	0.047	775	466	242	1.93	2.90	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.294	671	1255	[illegible]	22.5	0.041	578	425	281	1.51	2.74	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	55	0.144	425	960	0.093	13	0.035	383	387	197	1.96	2.57	1.39
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	11.0	565	0.305	2420	1500	0.355	125	0.064	1960	613	429	1.43	3.70	1.36
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4.9	425	0.256	1910	1530	0.294	55	0.057	1570	556	390	1.43	3.46	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	4.1	244	0.223	1364	1445	0.248	44	0.049	1154	475	328	1.45	3.19	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7.4	144	[illegible]	984	1250	0.161	29	0.039	769	376	254	1.48	2.94	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5.0	130	0.234	685	1160	0.124	16	0.029	571	374	298	1.25	2.79	
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	5.0	[illegible]	0.162	430	1060	0.065	11	0.030	381	373	206	1.81	2.63	1.39

during the entire throttling and mixture ratio excursion. The data illustrates how the oxidizer stiffness, varies by a factor of only 2.0 to 1 because of the heat addition to the oxidizer (gas/gas injection), rather than the 4.5 to 1 factor which would be experienced without heat addition (gas/liquid injection). In order to experience the same minimum oxidizer stiffness, with liquid injection, the full thrust injector pressure drop would need to be more than twice as great. The resulting oxidizer injector inlet pressure and pump discharge pressures would need to be 600 to 700 psi greater or a 25 to 30 percent increase. A savings in injector and oxidizer delivery system weight is thus obtained with the selected injector concept. Oxidizer pump maximum shaft power requirements are reduced by approximately 25 percent as well.

Figure 154 illustrates graphically the oxidizer density within the injection vane as function of thrust level and axial position within the vane. The vanes are designed such that the oxidizer is heated to temperatures above the critical temperature at all thrust levels. The injected oxidizer is a supercritical fluid, at chamber pressures above 740 psia (approximately 40% thrust), and gas at subcritical pressures. As shown, shorter axial vane lengths would allow two phase injection at minimum thrust levels. This condition is specifically avoided in order to preserve reasonable performance at deep throttling conditions. The baffle vanes, being longer provide additional superheating to the oxidizer; the internal channel dimensions differ slightly from the other fifty six vanes to provide the same flow rate per element. It is expected that slight variations in heat transfer to the oxidizer will, in any case, occur from vane to vane as well as within a given vane. Such variations will cause oxidizer density variations to occur from element to element. Since all elements are fed from a single source to a common chamber, local oxidizer density variations will cause variations in oxidizer flow rate from element to element. Such flow variation causes local mixture ratio shifts from the mean, and results in overall thrust chamber performance loss. Figure 155 shows the magnitude of the mixture ratio distribution (MRD) loss as a function of local fuel rich mixture ratio and the percent of elements operating at the fuel rich MR. In order to minimize MRD losses it is necessary to fabricate accurate and repeatable injector hardware. The photo-etch and braze fabrication method is outstanding in its repeatability, and its accuracy is especially good with copper platelets. If two phase injection were allowed at any thrust level, however, the MRD losses would be significant. Figure 154 shows how rapidly the oxidizer density changes with heat input near the critical point or under the dome (region of two phase co-existence). The selected injector design minimizes this problem by providing sufficient superheat to the oxygen to allow a 20% change in heat transfer without resulting in two phase injection at 20% thrust. This superheat is also useful in preventing two phase injection at deep throttling conditions caused by any non-uniformity in oxygen injection temperature, although this eventuality is minimized by the presence of a mixing plenum between the heat transfer passages and the oxidizer injection orifices. Additional superheat would reduce the likelihood of MRD performance losses at minimum thrust levels, but increases the weight and axial injector length, thereby decreasing the full thrust performance.

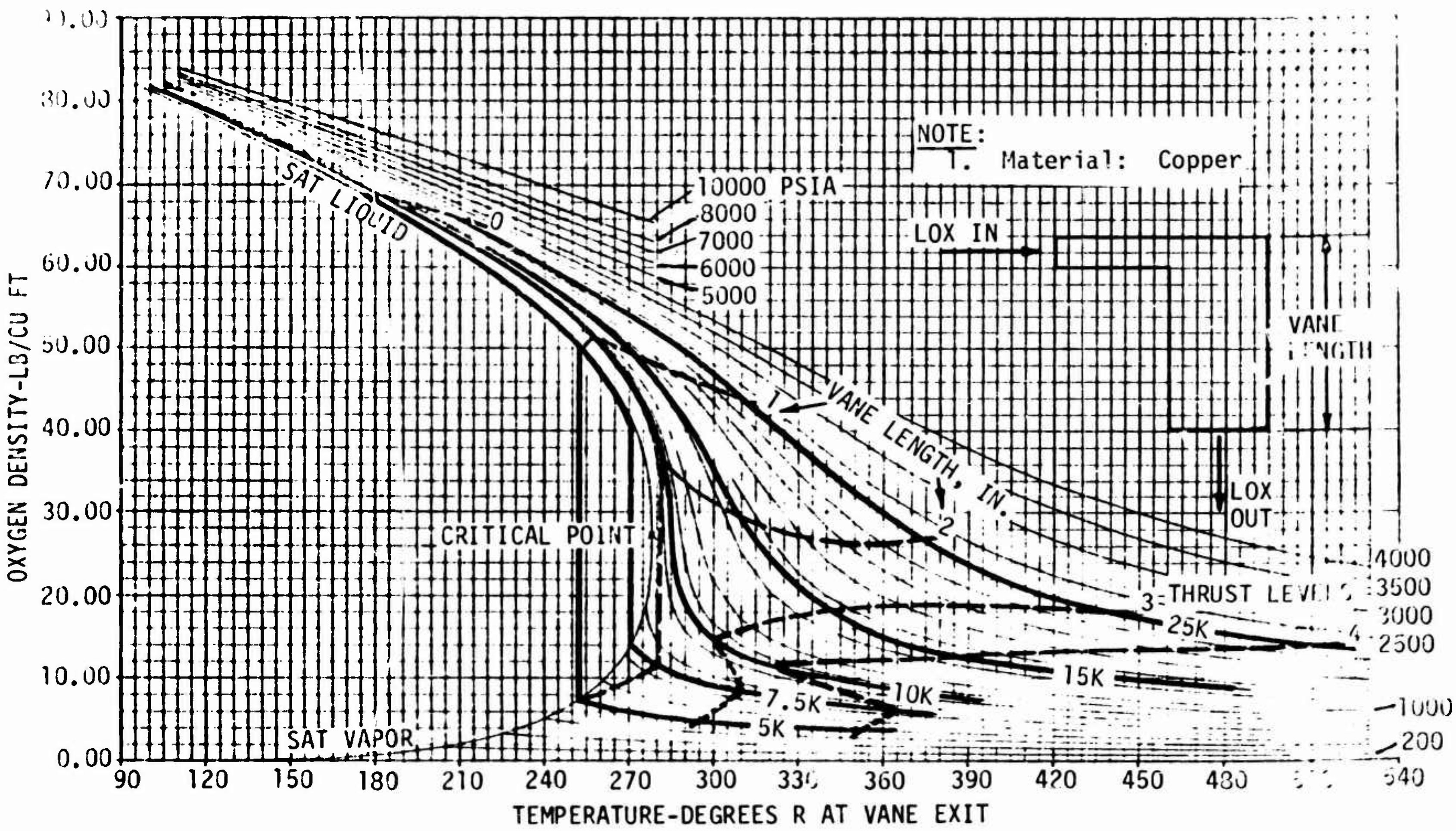


Figure 154. Effect of Vane Length on Oxidizer Discharge at Varying Thrust

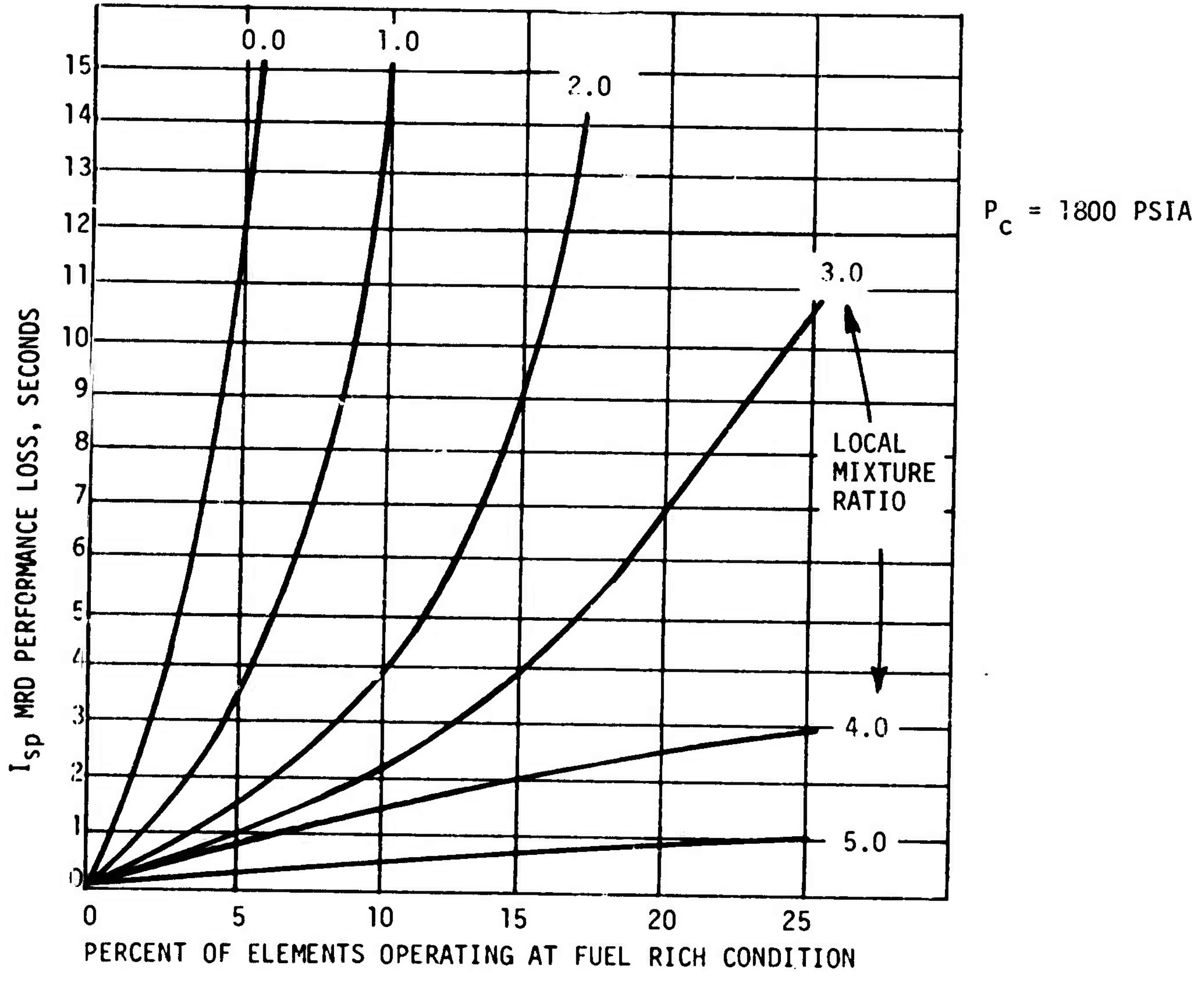


Figure 155. MRS Performance for O_2/H_2 TCA

The MRD performance loss problem is not unique to the selected injector design. Any vane type injector required to provide intimate mixing of fluids whose inlet temperatures vary by approximately 1500°F will transfer heat. In so doing, the aforementioned density variations will develop, especially considering the difference in manifold lengths which feed various oxidizer elements. In a fixed thrust engine design, appropriate adjustments can be made in manifold flow areas to offset otherwise unequal total heat transfer per pound of oxygen between short and long manifolds. In throttleable designs, however, variations in film coefficients caused by widely varying fluid velocities alter the thermal characteristics of the injector, and MRD losses must be encountered at some thrust levels. The philosophy of the selected injector is to take advantage of these heat transfer characteristic changes with thrust level, rather than to try to design around them.

It is noteworthy that at subcritical operating pressures, the gas/gas injection scheme eliminates the propellant varporization time lag from performance and combustion stability considerations. Propellant mixing is the sole major consideration. The impinging elements from fans whose tips penetrate the lower density FRHG force the mixing process to occur quickly (in a short chamber length). This is especially true with the selected injector design because of the small thrust per element value (35 lbf/element) and the high oxidizer injection momentum, compared to those obtainable with liquid oxidizer injection. Low oxidizer density (gaseous) injection requires a larger total orifice flow area for a given pressure drop and total flow rate than does high density (liquid) injection (see equation 4 below). This allows a greater number of elements to be utilized containing minimum sized orifices (orifice size determined by oxidizer contaminate size and/or fabrication limits). The greater the number of elements the more rapid the mixing will be, because the mixing distance is decreased. In addition, the oxidizer injection momentum is increased (see equations 5 and 6), and is available to more completely penetrate the FRHG. The following equations illustrate these relationships.

(1) $\Delta P = K \rho v^2$

(2) $w = \rho A v$

(3) $M = wv$

where:
ΔP = orifice pressure drop
ρ = oxidizer density
V = injection velocity
w = injection flow rate (total)
A = injection area (total)
M = oxidizer injection momentum
K = constant

Therefore:

(4) $A = w\sqrt{\frac{K}{\rho \Delta P}}$, and

(5) $M = \frac{\rho A \Delta P}{K}$ or (6) $M = w\sqrt{\frac{\Delta P}{\rho K}}$

The selected injector design can supply high Energy Release Efficiency (ERE) in a short thrust chamber because of its high

oxidizer momentum in conjunction with a small thrust per element impinging element pattern, which does not depend upon turbulent transport mixing in the chamber, but rather on rapid mechanical mixing across small distances near the injector face. Also, it has been indicated that the other major performance efficiency factor, MRD is promoted by accurate fabrication methods and sufficient oxidizer superheat to minimize oxidizer density and oxidizer flowrate variations from element to element. Small thrust per element patterns also aid combustion performance by minimizing those MRD losses caused by mixture variations within an element stream tube. For example, during laboratory testing of the ALRC SSME main injector pattern, it was found that the mixture ratio variation along an injector radius 1-in. from the face was from 5.6 to 6.4 with a 385 lbf/element pattern, but only 5.85 to 6.15 with a 113 lbf/element pattern. The selected element size of 35 lbf/element is expected to virtually eliminate this type of MRD performance loss.

Injector Fab Methods

The advantage of the vane injector concept is the capability and ease of making design changes to control the oxygen mass distribution. This capability was demonstrated during the ALRC SSME Phase B program. Examples include the control of the oxygen distribution near the injector wall by canting the outer two doublet elements, control of the oxygen distribution along the vane length by providing uniformly increasing orifice injection area to compensate for variations in O_2 injection density, and increasing the injection area of the last element of each vane to provide a uniform O_2 mass flux at the intersection of the vane "pie" sections. These design changes did not require new tooling or costly manufacturing changes, but merely required new vane drawings and art work from which the vane parts were etched.

The main injector vanes are a brazed assembly of two brazed subassemblies, manifold vanes and injection vanes. The injection vanes are composed of a furnace brazed stack of OFHC copper "platelets." The platelets are copper sheets which have had grooves and/or holes photo-etched into their surfaces, such that, when properly assembled, they form a plate containing a series of accurately located and sized internal flow channels. Transverse flow channels in the forward portion of the injection vane effect final oxygen flow distribution prior to entry into the heat exchanger section of the vane. The distribution channels are formed by the forward-most internal walls of the injector vane, and by the outer edges of the manifold vane, which slips into the injection vane, as does a knife into a scabbard. The major portion of the injector vane contains axial flow passages in which heat is transferred from the surrounding high velocity FRHC into the high velocity oxygen. At the trailing edge of the injector vane is an inlet plenum region which feeds a series of orifices forming impinging oxidizer doublet and showerhead injection elements. The arrangement of vanes in the injector assembly is shown in Figure 152. This figure also shows how the injection vanes penetrate the slotted injector body (ring), and how the oxidizer manifold vanes connect the slotted oxidizer torus with the injection vanes.

The manifold vanes are a furnace brazed stack of stainless steel platelets, fabricated in a manner similar to the injection vanes. The manifold vanes intersect the oxygen inlet torus at milled and brazed slotted joints in order to support the torus and to transfer the oxidizer from the torus through internal flow passages to the injection vane distribution passages. After flow testing the manifold vanes are brazed to the injector vanes at the radially outer tip of the injector vanes only, to effect a fluid seal and to mechanically restrain the distribution vanes at their midpoint. The bifurated inboard portion of the manifold vanes do not contact the injector vane, but are isolated from them by the oxygen, which is allowed to escape into the injection vane distribution passages uniformly along the forward edges of both manifold vane legs. This method of construction allows isothermal distribution of liquid oxygen into the hot injection vane, and prevents undesirable thermal gradients in the injector hardware.

The oxygen torus manifold and inlet line is a formed and welded sheet metal subassembly. The Armco 22-13-5 manifold is completed when the machined inlet flange has been welded onto the inlet stand pipe and the sixty-four vane receiving slots have been machined into its forward side.

The injector body is machined in eight segments from Armco 22-13-5 plate. Each segment contains eight slits which receive the copper injection vanes for brazing. After brazing the eight segments are welded together to form a complete housing/injector vane subassembly. Following this, the manifold vanes are slipped into the injection vanes, and the oxidizer torus manifold is positioned around and between the manifold vanes. A final furnace braze and seal groove machining operation completes the injector assembly. The outer housing of the hot gas inlet manifold is machined from a forging of Armco 22-13-5 material. The internal FRHG liner is a TIG and resistance welded subassembly of formed 22-13-5 sheet stock, and machined distribution plates of the same material. The liner is slipped inside of the housing such that the parallel liner inlets protrude through the inlet holes in the side of the housing. The two parts are then TIG welded together using split thermal standoff pipes, as shown in Figure 151. A similar welded attachment method is used at the igniter/housing interface opposite the fuel inlets.

Combustion Component Low Cycle Fatigue Life

The engine requirement which most greatly affects the major combustion component designs is the thermal cycle life requirements. Table XXXI lists the fatigue life margins predicted for all combustion components. The hardware failure mode encountered with long life combustion components is metal fatigue cracking. This type of failure occurs when the metal is repeatedly strain cycled beyond its elastic limit, such that permanent deformation occurs without single load cycle material fracture. Metals may be strained in two ways, by application of a load by an external member, or by being restrained when heated or cooled. Further, the restraint may be provided by an external structure, or internally by the metal itself. Plastic strain caused by external load application is easily reduced to the

TABLE XXXI

FATIGUE LIFE AND MAINTENANCE SCHEDULES FOR COMBUSTION COMPONENTS

Summary of Structural Analysis for 50 hr Life

Component	Fatigue Life Capability	Expendible Mode 10 Cycles/O'Haul		Reduced Reqmnts 60 Cycles/O'Haul		Baseline Design 300 Cycles/O'Haul		Increased Reqmnts 600 Cycles/O'Haul	
		Replace Period	Factor of Safety	Replace Period	Factor of Safety	Replace Period	Factor of Safety	Replace Period	Factor of Safety
Main Combustion Chamber Zirconium Copper Liner T_{max} = 900°F	1050	None- 50 Cycles	21.0	None- 300 Cycles	3.50	Each O'Haul- 300 Cycles	3.50	Second Refurb. 240 Cycles	4.37
Regen-Cooled Nozzle Tubes Armco 22-13-5 Stainless Steel Tubing 707°F max	10,200	None 50 Cycles	High	None- 300 Cycles	34.00	None- 1500 Cycles	6.80	4th O'Haul 2400 Cycles	4.25
Main Injector Vanes- OFHC Copper Outer Platelets T_{max} = 932°F	9800 @ 50 hr (1)	None- 50 Cycles	High	None- 300 Cycles	32.67	None- 1500 Cycles	6.53	4th O'Haul 2400 Cycles	4.08
Preburner U-Tube Liner Armco 22-13-5 Stainless T_{max} = 765°F	8800	None- 50 Cycles	High	None- 300 Cycles	29.33	None- 1500 Cycles	5.87	3rd O'Haul 1800 Cycles	4.88
Nozzle Skirt Extension AGCarb Material T_{max} = 2900°F	$>10^6$ (2)	None- 50 Cycles	High	None- 300 Cycles	High	None- 1500 Cycles	High	None- 3000 Cycles	High

(1) Only component affected by creep damage during extended duration operation

(2) Not sensitive to cyclic operation, total duration limited by ablation/erosion.

elastic range by increasing the load carrying section, or by reducing the local load through detail configuration changes (e.g., elimination of large stress concentrations, etc.). Strain caused by restraint of a heated or cooled object is, however, less easily reduced, because the material itself changes volume, and an increase in section normally has no effect on the strain level. Rather, the amount of restraint, the thermal differential, and the material expansion and fatigue properties are the important factors.

When a metallic object is heated it is not necessarily strained. A uniformly heated flat plate will, if unrestrained by any structure, not become strained or stressed, because its physical dimensions are allowed to change as required. If the plate is heated between two attached and immovable end structures, a compressive strain will be developed in the plate in a direction of the restraints. Similarly, the unrestrained flat plate which is slowly heated with a linear thermal gradient will not be strained if the material has a linear coefficient of expansion, because the required growth corresponds to free rotation of the elements within the plate. Application of a non-linear thermal gradient to the plate, however, will cause material strain, because of internal restraint. The cooler section of the plate restrains the hot portion and visa-versa through the establishment of internal shear stresses. Thus, the hot section under goes compressive strain, and the cold portion tensile strain. Because the material modulus and other properties change with temperature, it is often found that the heated portion will strain a greater amount than the cold portion, so that, normally, compressive strain is the item of concern. Non-linear thermal gradients occur in open structures during rapid thermal transients and also under steady conditions when local heat transfer alters the thermal profile of the part.

Closed structures, such as rings, cylinders, toruses, etc., like open structures, such as flat plates, suffer no strain if heated uniformly without external restraint. However, the application of any thermal gradient to the part induces internal restraint and strains the material. An unrestrained cylinder, for example, which is heated internally and cooled externally undergoes compressive straining internally, and tensile strain in the outer fibers, in the hoop and axial directions. The following table is a summary of the foregoing discussion, relating physical and thermal conditions to the development of strain in a structure.

	Unrestrained Externally			Restrained Externally		
Material Strained	Uniform Temp Change	Linear Thermal Gradient	Non-linear Thermal Gradient	Uniform Temp Change	Linear Thermal Gradient	Non-linear Thermal Gradient
Open Structure	No	No	Yes	Yes	Yes	Yes
Closed Structure	No	Yes	Yes	Yes	Yes	Yes

Thermally induced material strain will not cause fatigue failure if the strain levels are within the material elastic range at the local material operating temperature. However, the total strain at any location of interest is the factor which determines the fatigue life of a given material at a given temperature. The strains in all three geometric coordinates must be algebraically summed to determine the total strain of the material, and is quoted in percent (e.g., 100 x in./in.). The life of a part is determined by its local area which has the shortest fatigue life, so that the detailed geometry of a part is an important fatigue life factor. Any part whose section changes abruptly is apt to suffer an early fatigue failure because of strain concentration in the thin section. In addition, parts with great inherent stiffness develop strain more readily than do flexible ones. Flexible members, when heated may elastically warp, thus relieving some of the strain energy.

Some materials have better fatigue resistance than others, although in almost every case the ratio of strain cycles to failure vs total strain is dependent upon the material temperature, as well as the material. Also, if the working temperature is high enough and the strain is held for a significant period of time during each load cycle, creep rupture damage can occur. The creep damage factor is additive to the strain magnitude in reducing fatigue life.

Fatigue characteristics are not generally listed throughout materials literature, especially at elevated temperatures. Therefore, a materials testing program is essential to good component life design.

Heated structures may be actively cooled if a cold heat sink is attached to the structure at some location remote from the heated area. The cold sink may be a cooler structure or a cool fluid passage. In any case, the flow of heat through a component requires the existence of a thermal gradient. In most cases, this causes thermally induced strain. In general, the larger the gradient, the larger the strain, so that individual components of a larger assembly should be located such that they contact fluids or solid structures of similar temperatures. This implies that for maximum fatigue life, individual components should not be cooled by conduction of heat through the structure, but rather allowed to operate at adiabatic conditions, if possible. Where conduction cooling is used, the coolant temperature should not be significantly below the maximum temperature of the part. This implies that conduction cooling is most useful in low heat flux regions, where heat can be removed from a part rapidly enough without a large temperature differential between the source and the sink. It also implies that cold structures which <u>are</u> conduction cooled to prevent melting or gross loss of material properties will necessarily develop significant strain and encounter fatigue life limitations, because the coolant must be cold relative to the heated area in order to be effective. The fatigue life of such components can be increased if some form of barrier cooling is used to reduce the heat flux to and temperature of the heated portion of the structure. Mechanical shields may be used if material melting is not the problem, but rather

loss of strength. The shield should not be conduction cooled lest the fatigue problem be merely transferred to the shield. Rather, the shield should be uncooled, and freed from mechanical loads, so that its poor mechanical properties are acceptable. Fluid barrier cooling is a very effective method of increasing component fatigue life, for it prevents the structure from being subjected to the hostile thermal environment, and avoids some or all of the conduction-cooling induced plastic material strain.

Based on the foregoing discussion, the following guidelines for extended combustion component fatigue life are established:

(1) Minimize external structural restraint.

(2) Where possible, utilize open structures rather than closed.

(3) Minimize non-linear thermal gradients in all structures.

(4) Design for low temperature differentials in single piece closed or restrained structures.

(5) Minimize thermal shock by minimizing thermal transient rates, and by using thin and/or high conductivity material.

(6) Minimize the rigidity of a structure to allow elastic bending to relieve regions of highest strain.

(7) Avoid rapid changes in section in a single part to avoid strain concentrations.

(8) Select a material which has the good inherent fatigue resistance at the expected operating temperatures.

(9) Select a material not susceptible to creep rupture damage at the expected operating temperatures.

(10) Conduction cooled closed structures only in low heat flux, low temperature environments. Use large L/d design and low conductivity material to reduce heat flux and reduce thermal gradient (°F/in.).

(11) Use load balanced uncooled shields to protect structures where heat source temperature is below shield material melting temperature.

(12) Use fluid barrier cooling to protect structures where cooling would otherwise be needed to prevent melting.

During initial design phases each component must be reviewed with respect to fatigue life. Since major components are assemblies of individual structures, it is necessary to arrange the structures so that unwanted conduction cooling is not inadvertently caused by attachment of a heated structure directly to a cooled one. Similarly, major component interfaces must observe the same guideline. Therefore, it is necessary to establish an acceptable thermal network within each component and, in fact, throughout the entire engine/vehicle structure.

Minimum and maximum OOS engine life requirements are specific, while goals, such as performance, maintenance cost, weight, fabrication cost, etc., are not. This implies that fatigue life enhancement methods which reduce engine attractiveness in these other areas should be used sparingly, and only if the minimum life goals cannot otherwise be met, or if a large payoff in maintenance, development, or fabrication cost is realized. Those techniques which can be incorporated without penalty to other goals should, of course, be implemented. Where specific tradeoff factors are available, such as weight vs performance, an optimum configuration can be obtained with the expenditure of trade-study effort. In other cases, it is not possible to do this; for example, the performance vs cost tradeoffs require judgement. In parametric studies the tradeoff results are sufficient, while they are not when a point design is required. Since an "optimum" design is affected by the designer's assumptions, it is imperative that all relevant assumptions be listed. The following sections which justify the selected component designs, therefore, include listings of assumptions and specifications.

In a staged combustion engine, the main injector encounters severe thermal gradients, because the two inlet propellants to be mixed are hot turbine exhaust gases (FRHG), and cryogenic oxygen. In the case of the subject engine, the propellant temperature differential amounts to approximately 1500°F. This single fact indicates that the design of long life injector and inlet hot gas manifold assembly requires application of the above fatigue life guidelines in order to minimize the maintenance requirements of these two components.

The actual injection element is the only place in the injector where the two propellants must be immediately adjacent, and this is the location of primary concern. In the subject design, this component is the injection vane.

Each vane is externally heated on both faces by the FRHG flowing at high velocity along its entire length, and cooled internally by the oxygen flowing axially within it. The resultant vane cross-section thermal gradient must be minimized, in order to maximize the vane fatigue life. Since the vane is hollow, it must be considered a closed structure in the coordinate direction defined by this thickness. The temperature differential across the vane wall is less than the difference between the FRHG and oxygen bulk temperature by the amount of the sum of the FRHG boundary layer ΔT. The size of the three ΔT's, vane wall, FRHG boundary layer, and oxidizer boundary layer are dependent on the local heat flux, which in turn are determined by the two fluid boundary film coefficients, and the wall thermal resistance. Since the vane is a heat exchanger; it is necessary to strike a proper balance between high heat flux to effect a compact injector, and low heat flux to reduce wall thermal gradients. The use of a low thermal resistance wall material in the selected vane design allowed a reasonably high heat flux to be obtained in conjunction with an acceptable wall ΔT. A vane length of 3.0-in. is sufficient to superheat the oxidizer at 20% thrust operation, the condition when the FRHG temperature and fuel and oxidizer velocities and film coefficients are minimum, so that minimum superheat is obtained. At higher thrust levels the injection temperatures are higher; in spite of the increased oxidizer flowrate. At full thrust, an oxidizer temperature rise of approximately 300°F has occurred within the vane, from inlet to injection plane (see Figure 154). A corresponding decrease in FRHG temperature of somewhat more than 200°F has also been effected. Therefore, in the region of the vane tip, the temperature differential between the two propellants has been reduced to approximately 1000°F. This is a very desirable feature of the heat exchanger injector. The liquid and gas film coefficients are very difficult to control at the injection face, because of the need to establish injection velocities on bases other than fatigue life, e.g., feed system hydraulic stiffness, propellant chamber mixing criteria, etc. Therefore, the heat flux, and wall ΔT cannot be well controlled in the area. In addition, to the FRHG influenced heat input and corrective heat transfer at the actual vane trailing edge, the high temperature secondary combustion in the thrust chamber raises the vane external temperature. In this region, the FRHG actually becomes a barrier coolant for the vane tip.

The baffle vane tips are exposed to a somewhat more severe thermal environment than are the other vanes, because they extend into the combustion zone by 0.5-in. The FRHG should act as barrier coolant flow along their sides. The barrier mixture ratio will vary from 0.8 at the

injector face to a value significantly less than 3.0 at the baffle tip. The reasons for this are that the wall barrier mixture ratio should remain 0.8 for a short distance, possibly as far as 0.5-in. In any case, the maximum local core mixture is 3.0, because the baffle vane has not yet been injected. Therefore, within 0.5-in. the wall mixture ratio will be much less than 3.0, because of the necessary presence of a transverse mixture ratio gradient from 3.0 down to the unknown wall barrier M.R. An injector development problem will be to determine the exact injection pattern and local flowrates needed to obtain the desired fatigue life safety margin.

Significant MRD losses are not anticipated as a result of the presence of baffles. The local core M.R. of 3.0 adjacent to the baffle will be corrected to the nominal core M.R. just downstream of the active, oxidizer injecting baffle tip. Since the oxidizer is warmer at the baffle tip than at the other vane tips, its injectant will contain greater momentum. Therefore, it is expected that the baffle injected oxidizer will be able to penetrate the adjacent gas effectively and prevent the accumulation of additional MRD losses.

It is noteworthy that the selected baffle tip design is both regeneratively-cooled by the oxidizer and barrier cooled by the FRHG, and, therefore, follows the same design philosophy presented in the following section (III,B,2,b) for thrust chamber cooling.

The remainder of the total strain imposed upon the vane wall is a result of axial thermal gradient in the vane. The vane is constructed as a cantilevered flat plate, attached to the injector body only, and at only one point. Because of this, a large thermal differential of approximately 800°F can be accommodated without contributing significantly to the total strain. The axial gradient can be linearized to a great extent by careful oxidizer channel sizing vs axial station, and since the vane is a largely unrestrained plate, the strain levels will be low. The vane axial thermal differential is taken over a 2.5-in. length, so the thermal gradient is less than it would be for a shorter component. The forward portion of the vane is necessarily connected to the injector body. It is, therefore, required to match the temperature of the body rather closely with the forward vane end in order to avoid a sharp radial thermal gradient in the vane. This was done by attaching the injector body to the copper thrust chamber, where the backside temperature is maximum, e.g., at the coolant outlet manifold. This manifold is located at the chamber forward end for this purpose; the local chamber temperature is approximately 0°F. Thus, the combined thermal differentials of the vane leading edge (radially, and the injector body, axially, is only about 500°F). Further, the injector body temperature of 200°F is an ideal heat sink mount for the hot gas inlet manifold outer structure. Heat flowing into the manifold structure through the thermal standoffs at the FRHG inlet lines, has an ultimate sink other than the gimbal block, e.g., the injector body. Further, the -250°F liquid oxidizer torus is thermally attached to the injector body through large L/d low conductivity,

stainless steel manifold vanes, which are brazed to the injector vanes near their joint with the injector body. The ΔT and heat flux associated with the manifold vanes are low, so that the vane life is great, and the oxidizer in the manifold is not boiled or prematurely heated.

The fatigue lives and maintenance schedules of all combustion components are listed in Table XXXI. The fatigue life of the injector is 9800 thermal cycles, which provides a safety factor of 6.5 over the entire engine life requirement of 1500 thermal cycles. The injector life could be increased by three methods:

(1) Increase injector axial length and face area (reduce heat flux through vane walls).

(2) Decrease throttling requirement.

(3) Expend development effort to determine if Zr-Cu properties could be retained in injector vane during fabrication sequence, to eliminate creep rupture damage during long duration firings.

A structural analysis of all combustion components forms a portion of Appendix C.

The FRHG and oxygen pressure schedules through the injector are listed in Table XXXII at maximum thrust, mixture ratio = 6 operation.

TABLE XXXII

MAIN INJECTOR PRESSURE SCHEDULE AT NOMINAL CONDITIONS FOR BARRIER COOLING

Fuel Rich Hot Gas				Oxidizer	
Injector Core		Barrier Coolant Circuit		Injector Core	
Loss	ΔP, psia	Loss	ΔP, psia	Loss	ΔP, psia
Inlet Manifold	40	Inlet Manifold	40	Manifold Vane, Total	70
Distribution Plate #1	20	Distribution Plate #1	20	Injector Vane, Inlet	15
Distribution Plate #2	20	Distribution Plate #2	20	Heating Channel Friction	145
Inter-Vane Friction Loss	60	Friction Loss	15	Orifice Inlet plenum	90
Injection Velocity	15	Injection Velocity	60	Injector Orifice	290
TOTAL LOSS	155		155		610

b. Thrust Chamber

The primary function of the thrust chamber is to contain high temperature and pressure reactants until their exothermic combustion process is complete, and then direct them efficiently to and through a sonic nozzle throat. In actual practice, the thrust chamber also comprises at least a portion of the expansion nozzle needed to extract the maximum performance from the combusted gases. Because of the high temperature, pressure, and velocity of the contained gases, a second function of the chamber is to provide for active internal wall cooling such that long duration engine runs may occur as many as 300 times or more and totaling at least 10 hours operation in a single chamber without component failure. A final function of the chamber is to serve as a structural attachment link between the injector and the nozzle assemblies.

Thrust Chamber Design Description

The thrust chamber is shown assembled with the nozzle in Figure 156, and its basic design specifications are listed in Table XXXIII. The chamber is regeneratively-cooled with hydrogen and its inner wall contains nontubular coolant passages, such that its inner surface is smooth. The convergent portion of the chamber, which extends from the forward portion of the injector to the throat is conical in shape with a half-angle of 7.5° and has a combustion zone contraction area ratio of 2.0. The distance from the propellant injection surface (injector face) to the throat station is 5.0-in. Because the chamber surrounds the injector for an axial distance of 2.0-in., the convergent chamber length is 7.0-in., and its overall contraction ratio is 2.3. The expansion nozzle portion of the thrust chamber extends to a nozzle area ratio of 6:1, at which point the nozzle is attached by a brazed joint. The chamber features a zirconium copper inner wall, which contains 121 longitudinal passages, which utilize 100% of the total engine hydrogen as coolant. The coolant channels are rectangular in cross section; they have continuously variable depth as well as various widths along their length to provide a nearly optimum configuration with regard to cooling effectiveness, chamber life, and engine performance. Slight concessions were made in the channel design, where great reductions in fabrication cost were obtainable. For this reason the gas-side wall thickness was held constant, and the channel width is stepped rather than tapered. A description of the channel geometry is shown in Figure 157. The hydrogen coolant is introduced in heated condition from the nozzle at the common interface. The coolant exits the chamber near the injector interface through a tapered toroidal collector manifold, from which it is ducted to the engine preburner.

The thrust chamber inner copper wall is externally supported to accept hoop, axial, and bending structural loads. The copper wall is wrapped with 0.100-in. diameter ARMCO 22-13-5 stainless steel wire, which is held in place by welding at each end to resist the chamber pressure induced hoop loads. A 0.050 in. thick ARMCO 22-13-5 conic frustrum is used to react thrust and gimballing applied loads. This "thrust cone" is welded to the coolant outlet manifold at its smaller, forward end and to the coolant

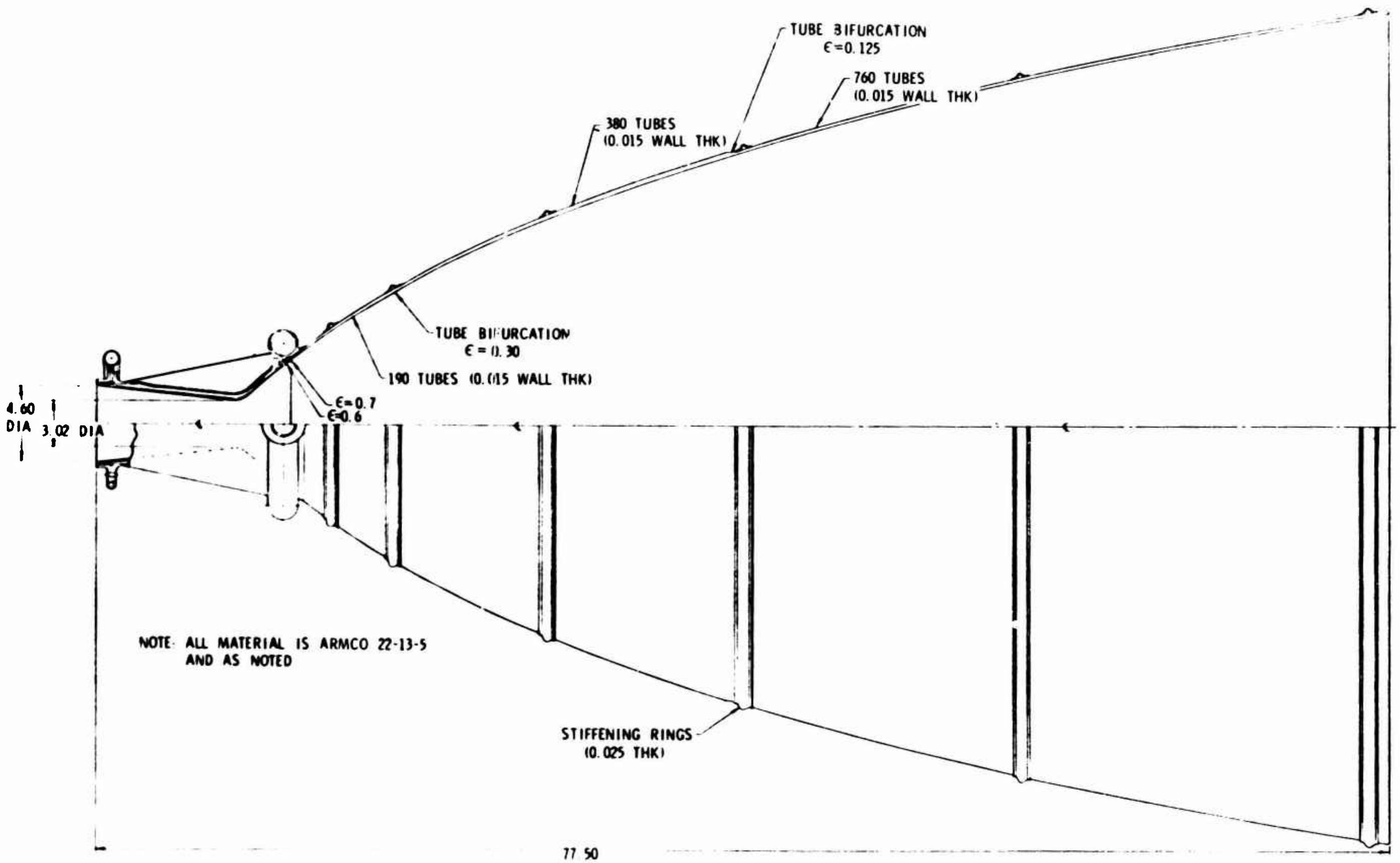


Figure 156. OOS Thrust Chamber and Nozzle Configuration

TABLE XXXIII

THRUST CHAMBER BASIC DESIGN SPECIFICATION

THROAT DIAMETER, IN.	3.02
CHAMBER CONTRACTION RATIO	
COMBUSTION ZONE	2.0
OVERALL INCL. INJECTOR	2.3
CHAMBER SHAPE	CONICAL
CHAMBER HALF-ANGLE, DEGREES	7.5
CHAMBER EXIT AREA RATIO	6.0
COMBUSTION LENGTH, L', IN.	6.5
OVERALL CHAMBER LENGTH, IN.	8.4
PRIMARY COOLING METHOD	HYDROGEN REGENERATIVE
FLOW SCHEME	SINGLE PASS, COUNTER FLOW
NUMBER AND TYPE OF COOLANT CHANNELS	121 RECTANGULAR

GAS-SIDE WALL THICKNESS, IN.	0.030 CONSTANT
CHANNEL DEPTH	CONTINUOUSLY VARIABLE
CHANNEL WIDTH	STEPPED, 3 WIDTHS
LAND/CHANNEL WIDTH RATIO AT THROAT	1.0
CHANNEL HEIGHT/WIDTH AT THROAT, IN./IN.	0.070/0.040
CHAMBER INNER WALL MATERIAL	ZIRCONIUM COPPER
HOOP STRESS SUPPORT METHOD	WIRE-WRAPPED
AXIAL LOAD SUPPORT METHOD	EXTERNAL CONICAL SHELL

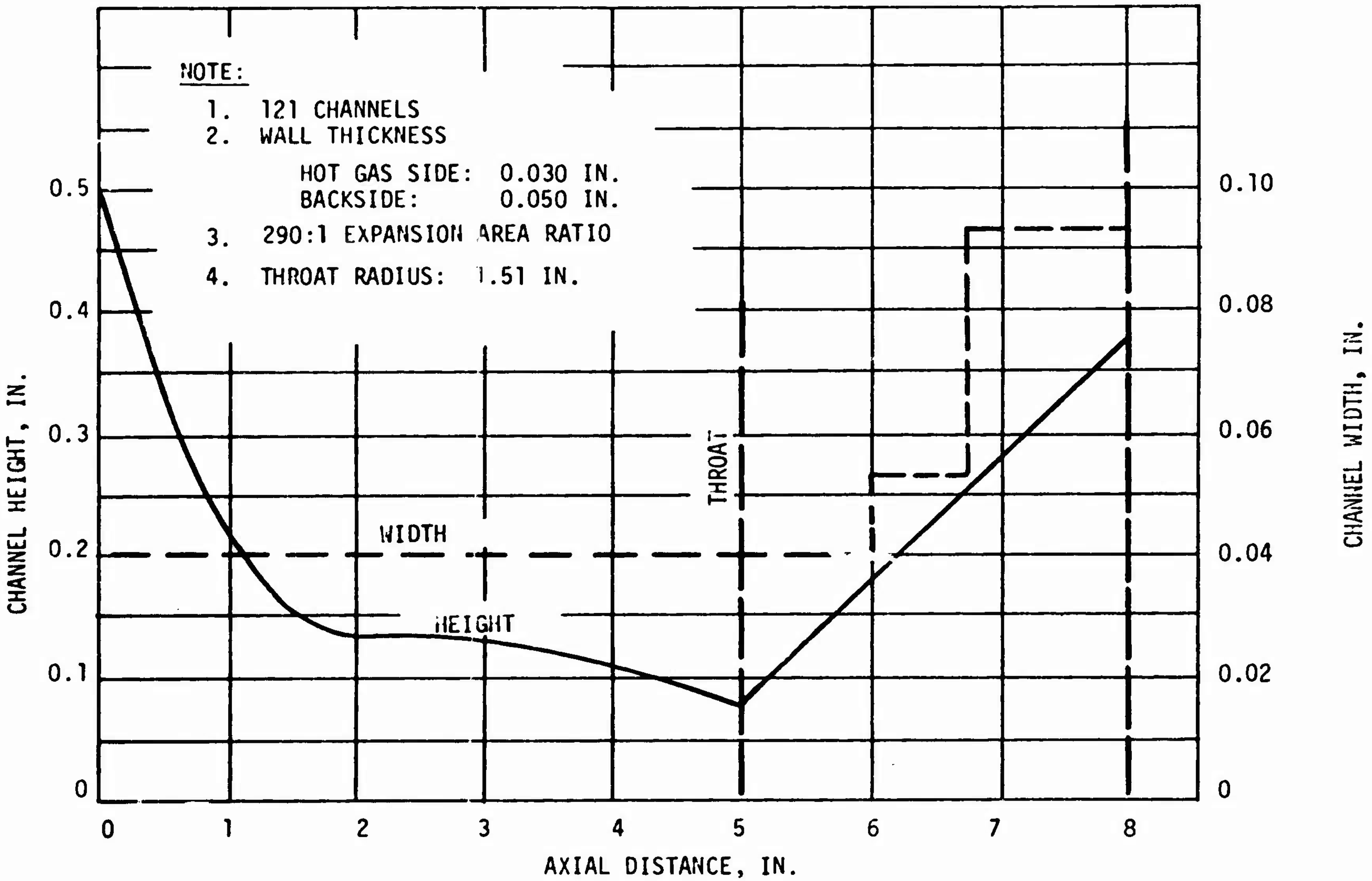


Figure 157. 25K Engine Copper Nozzle Coolant Channel Geometry

inlet manifold located at the forward end of the nozzle at its larger, aft, end. The cone also provides weld attachment locations for the chamber portion of the gimbal actuator devices.

The thrust chamber inner wall is fabricated using a single piece gas-side liner. The inner and outer contours are machined first, followed by milling coolant channels into the outside of the part. A second milling operation widens the channels, leaving an appropriate depth of the narrower grooves at each axial location on the chamber. Rectangular cross section wires are formed to match the chamber contour and inserted into the wider grooves, such that rectangular passages remain in the wall interior. The inner chamber is completed by furnace brazing the zirconium-copper channel closure wires in place, as well as the coolant outlet torus, and wire wrapping. A cross sectional view of the thrust chamber wall is shown in Figure 156. The thrust chamber weight, excluding the clevises is listed in Table XXIII, along with the other major combustion components.

Thrust Chamber Thermal Characteristics

The fuel system pressure drop which occurs in the thrust chamber coolant channels is in series with all other pressure drops in the fuel system. For a fixed chamber pressure engine, the required fuel pump discharge pressure is strongly affected by the coolant pressure drop in the thrust chamber wall. Less obvious, however, is the fact that the coolant pressure drop in the thrust chamber is affected by the pump discharge pressure. Figure 158 shows the effect, calculated for the Task I thrust chamber design at 100%, 20%, and also for 10% of maximum thrust level. Shown is that an increase in fuel pump discharge pressure (coolant inlet pressure) causes disproportionately large decreases in coolant pressure drops. This is caused by the change in volumetric flowrate with coolant pressure. Also shown is that the coolant bulk temperature rise through the chamber is not affected by inlet pressure, but that it is by thrust level. The coolant temperature rise is 100°F greater at 20% thrust than at 100% thrust. This increase in temperature rise is caused by the reduction in coolant flowrate relative to the total heat load from the chamber at throttled conditions. The chamber life is increased by throttling since the heat flux and radial wall thermal gradients are reduced during low chamber pressure operation.

The interesting point of the chamber coolant ΔP vs inlet pressure interaction described above is as follows:

(a) Thrust chamber low cycle fatigue life requirements dictate a family of thrust chamber gas-side ΔT_W wall temperatures, T_{wg}, and radial wall temperature differences illustrated in Figure 159.

(b) For a given coolant inlet temperature (dictated by tank fuel temperature, pump efficiency and discharge pressure), the chamber T_{wg} and ΔT_W are determined by chamber axial location, hot gas boundary layer, mixture ratio (temperature), chamber pressure, and coolant velocity. This relationship is shown for nominal conditions in Figure 160.

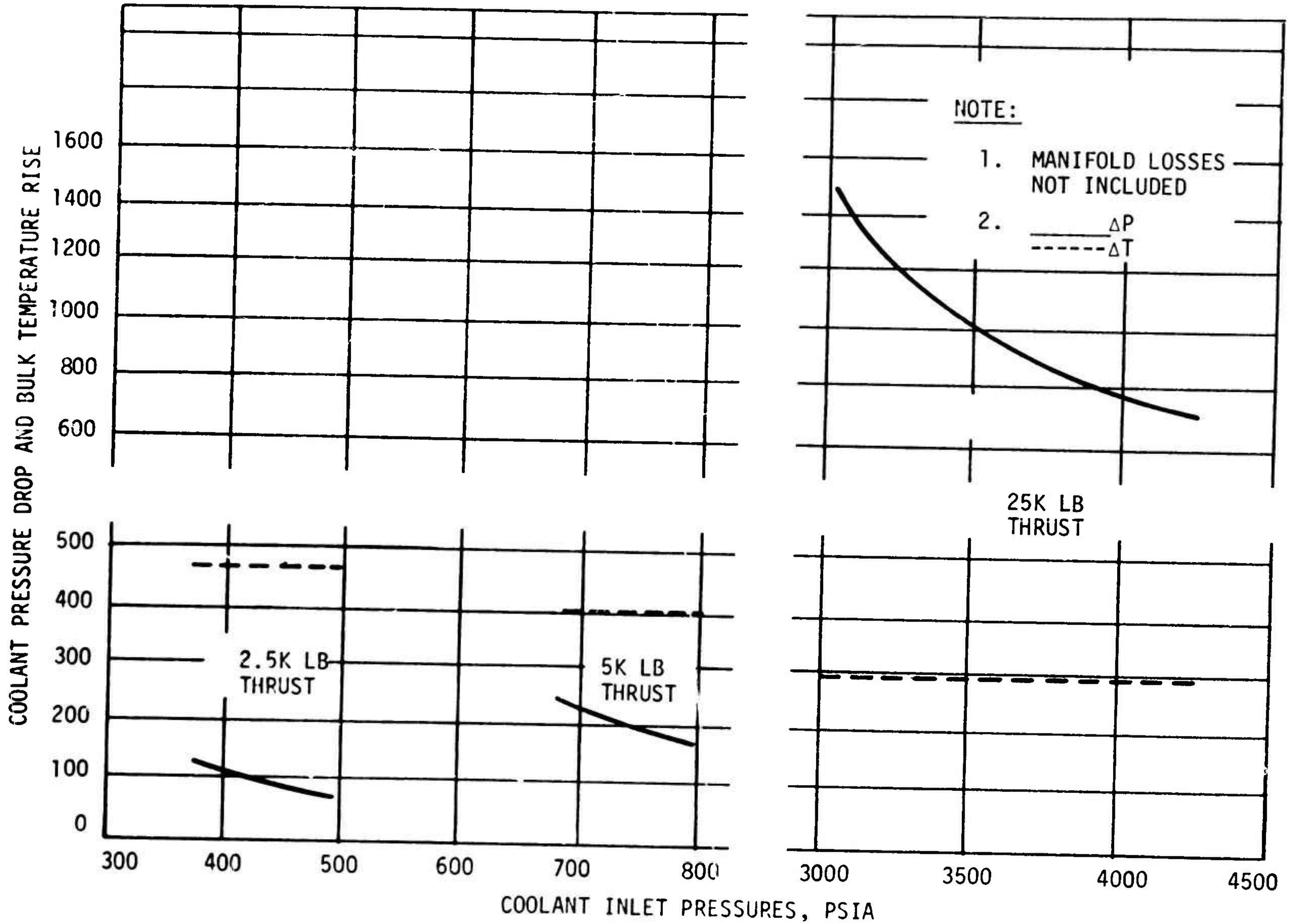


Figure 158. LH_2 COOLANT PRESSURE DROP AND BULK TEMPERATURE RISE FOR VARYING INLET PRESSURE

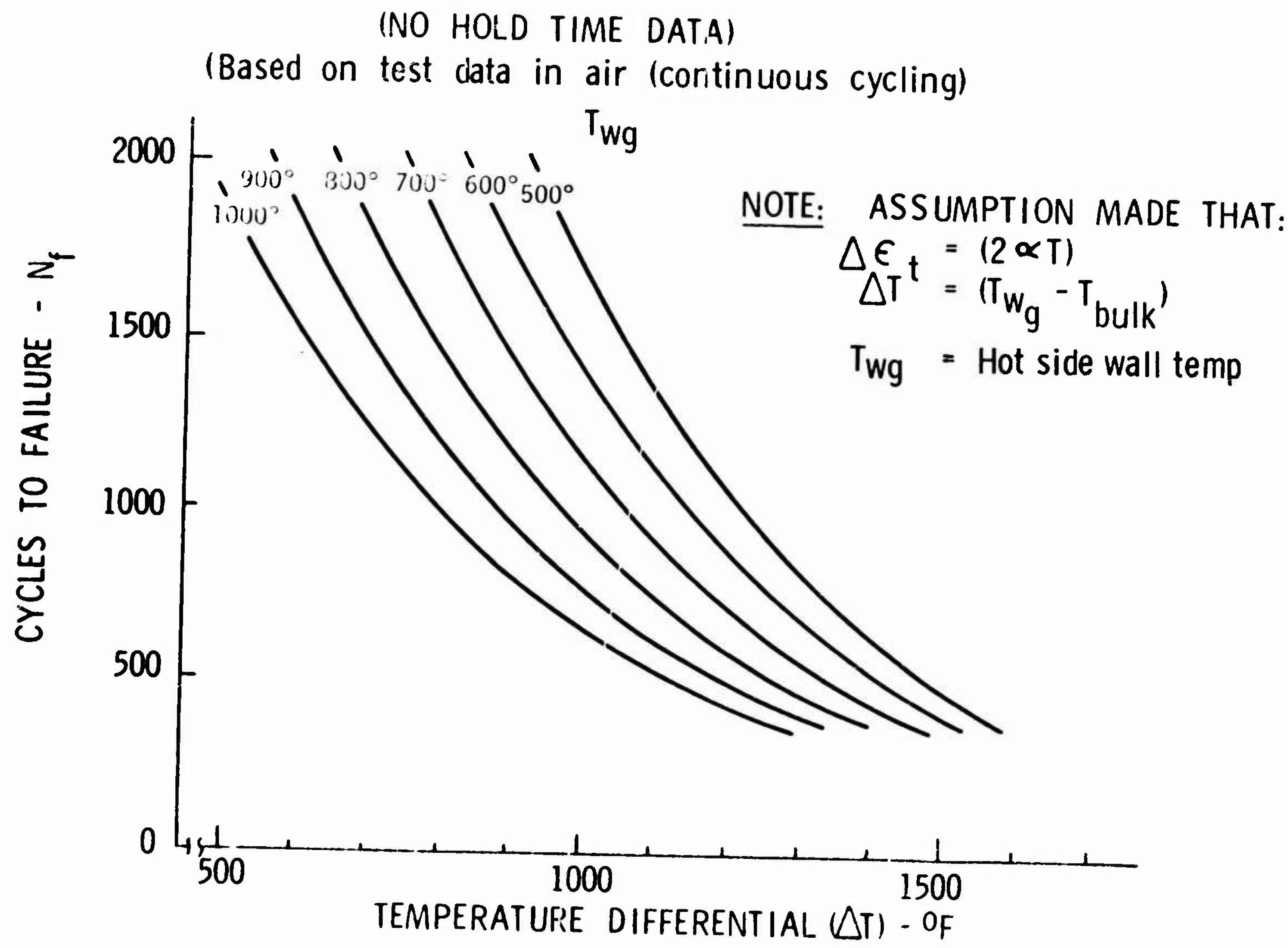


Figure 159. Zirconium Copper OOS Chamber Low Cycle Fatigue Life (LCF) Requirements

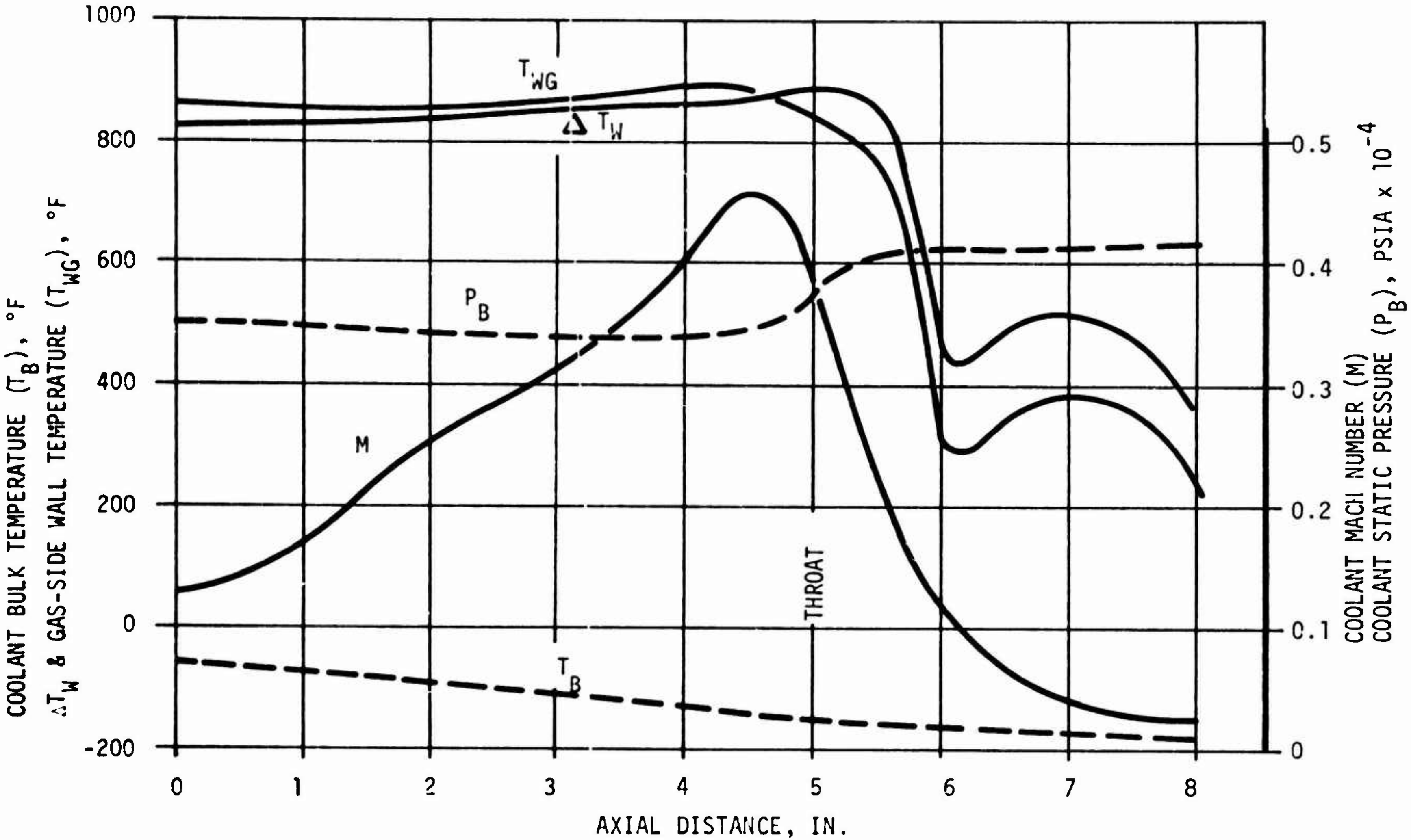


Figure 160. Copper Nozzle Design Parameter Profile

(c) To maximize engine performance in a given envelope, the chamber pressure must be maximized to maximize nozzle area ratio and minimize weight.

(d) Since heat fluxes are the greatest near the throat, and assuming a fixed gas boundary layer mixture ratio and coolant temperature, the T_{wg} and T_w (chamber life) are determined by chamber pressure and the coolant velocity at the throat of the chamber wall.

(e) Since high coolant velocities result in a low T_{wg} and T_w (long life), but also in large total pressure drops in the cooling passages, and are increased by chamber pressure increases, the fuel pump discharge pressure requirement rises very rapidly with increases in chamber pressure.

(f) Fortunately, increases in pump discharge pressure limit the rise rate of coolant pressure drop with chamber pressure, and therefore, limit its own required rise rate with chamber pressure.

(g) An iterative design process is required, and when completed, results in the highest chamber pressure design (and highest performance engine) consistent with chamber and turbopump life requirements as well as system power balance goals.

Thrust chamber thermal analyses referred to in this section are amplified in Appendix B.

The chamber operating conditions with the characteristic shown is presented in Figure 161 for throttleable conditions and in Figure 162 for off-mixture ratio operating at full thrust including major life capability pressure drop and chamber wall conditions. The data presented (Figure 162) is valid for a fixed chamber pressure operating under off design conditions contrary to Figure 163 representing rubber type thrust chamber designs.

(1) Thrust Chamber Concept Selection

Regenerative, transpiration, and gas-side barrier cooling were considered for use with the OOS thrust chamber. Both regenerative and transpiration cooling concepts have been successfully demonstrated at high chamber pressure: transpiration cooling in Contract AF 04(611)-10830 and AF 04(611)-11401, and regenerative cooling in AF 04(611)-67-C-0093. The results of detailed performance analyses performed at ALRC for the SSME design revealed that the regenerative cooling system provides a substantially higher engine performance potential. The indicated nominal degradation due to transpiration cooling of the thrust chamber was greater than 5 sec of specific impulse for large nozzle area ratio engines when compared to that of a regeneratively-cooled system. Subsequent comparisons of the transpiration cooling performance model with actual hydrogen transpiration-cooled chamber data obtained in Contract AF 04(611)-11401 showed good agreement.

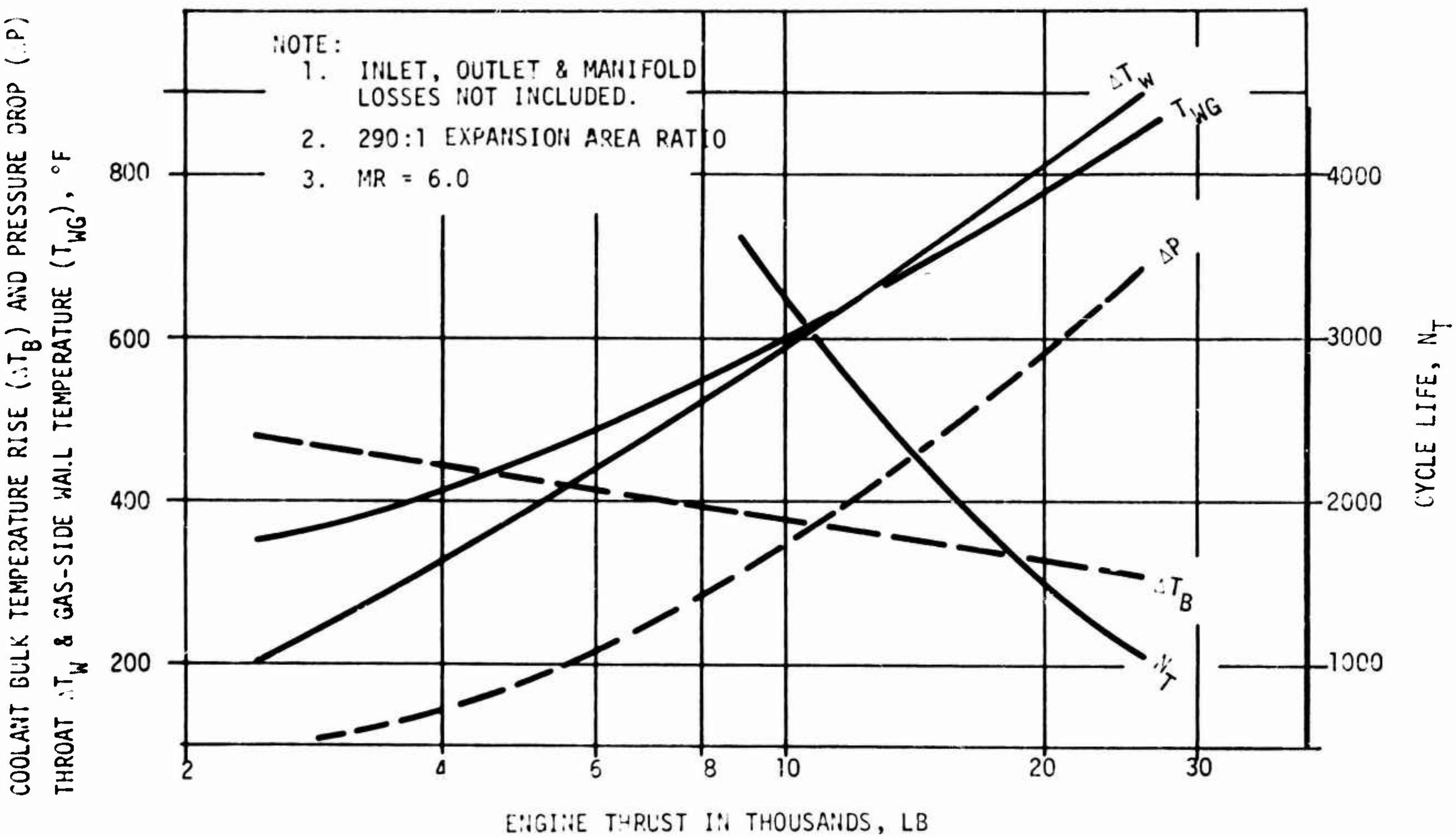


Figure 61. Copper Nozzle Effects of Throttling the 25K Engine

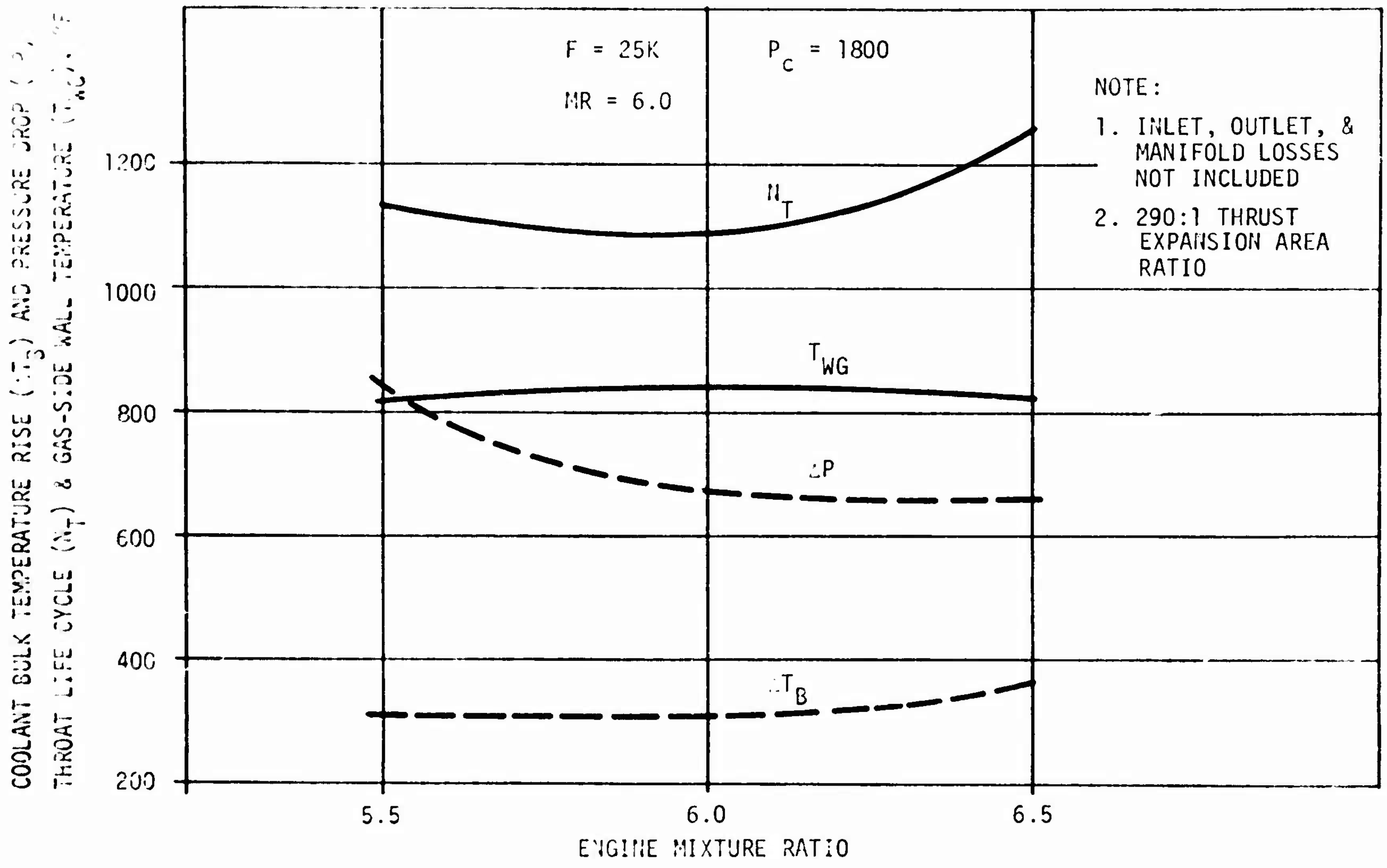


Figure 162. Copper Nozzle Effects of Mixture Ratio Rubber Engine Design Condition

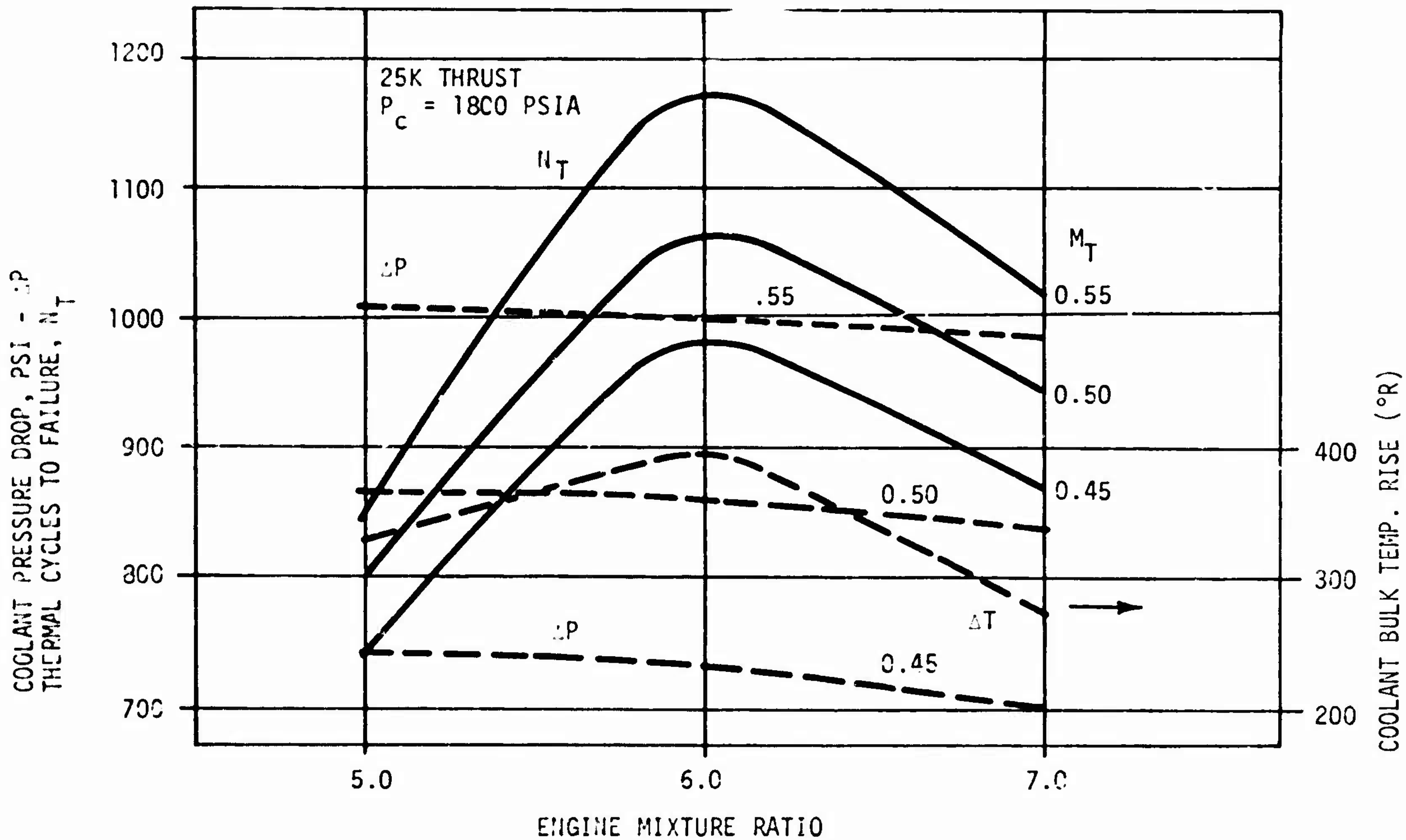


Figure 163. Coolant Pressure Drop and Bulk Temperature Rise vs Mixture Ratio and Throat Mach No. Point Design Engine - Off Design Conditions

Performance losses caused by transpiration cooling are largely caused by MRD losses induced because a portion of the hydrogen is not mixed with the oxygen at the injector face. In a small engine, such as the OOS, where the specific chamber surface area (chamber wall area per unit fuel weight flow) is large relative to an engine of the SSME thrust class, a larger transpiration bleed loss would be expected. In addition, transpiration-cooled chambers are heavier than corresponding regenerative components, because the chamber coolant manifolding and injectors required for effective transpiration cooling results in heavy specific wall weights ($lb/in.^{-2}$). On a payload basis, then, the expected loss in OOS engine performance would be well in excess of 5 sec of specific impulse, compared with a regeneratively-cooled chamber design.

The regenerative cooling system requires no coolant loss to the chamber, but it does require large total pressure losses through the coolant passages in order to provide effective cooling during high chamber pressure operation. As mentioned above, the cooling requirements are more severe for small engines than for large ones, so that all of the hydrogen must be used to adequately cool the chamber, and must be passed through the chamber in series with the turbines, injector and thrust chamber. The deleterious effects of bypassing some of the hydrogen is shown in Figure 164. Since all of the major fuel pressure drops are additive to the high chamber pressure, the required fuel pump discharge pressure is quite high, e.g., on the order of 2.5 times the chamber pressure.

(2) Thrust Chamber Barrier Cooling

Barrier cooling imposes an engine performance penalty for the same reason that transpiration cooling does; it induces MRD losses in the chamber. This concept was not reviewed as a primary chamber cooling scheme, but rather as a supplementary system. The barrier cooling scheme which appears attractive for use with the regeneratively-cooled chamber involves injection of pure FRHG at a temperature of approximately 1100°F in a thin sheet along the chamber wall from the injector periphery. As the cool gas moves aft along the chamber wall, it mixes with, combusts with, and is heated by the reaction products in the chamber core. Radial and axial temperature gradients develop in the boundary during steady operation, which provide an axial boundary mixture ratio and temperature profile along the thrust chamber wall, which begins (at 0.85, respectively) 1100°F at the injector and approaches the chamber gas conditions at remote downstream axial locations. The wall boundary mixture ratio (and temperature) at the chamber throat, where the highest wall heat flux, and thermal gradients, and minimum

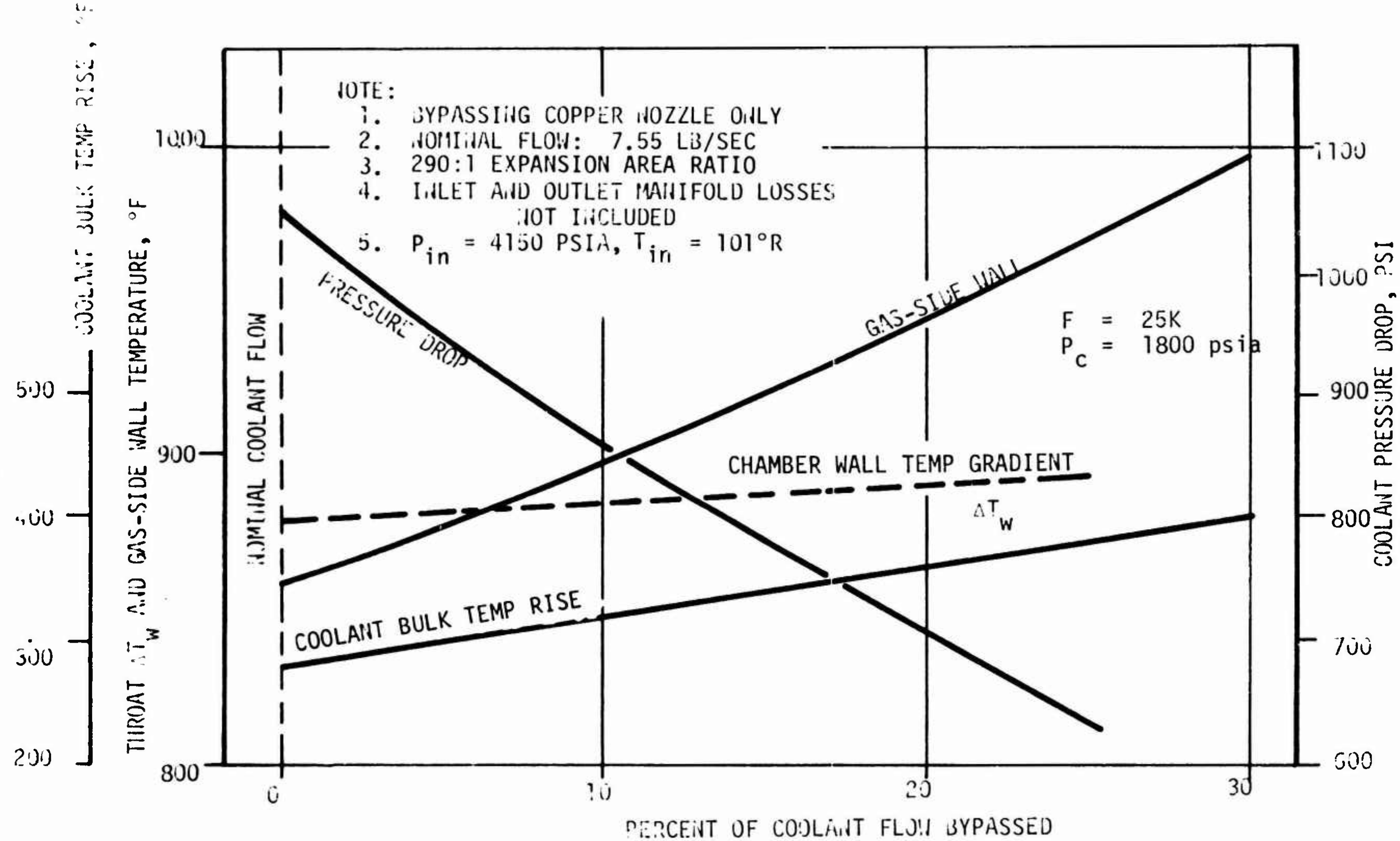


Figure 164. Zirconium Copper Nozzle Effects of Bypassing Coolant Flow

chamber life are normally encountered, is a function of several variables, but notably of the chamber length from injection point to throat plane (L'), the engine and barrier mixture ratios at the injection point, the chamber gas/coolant gas velocity ratio, and the amount of barrier coolant supplied. Because of the injector configuration, the chamber length required for high combustion efficiency is relatively short at 6.5-in. This causes throat barrier cooling to be more attractive than would otherwise be the case, because it reduces the required coolant injection flowrate and the resulting MRD performance losses.

Figure 165 shows the results of a heat transfer analysis conducted to determine the effect of wall barrier mixture ratio on thrust chamber throat low cycle fatigue life. This analysis showed that wall barrier mixture ratios below 3.5 significantly increases thrust chamber life. This data was generated by comparing the calculated gas-side wall temperature (T_{wg}) and radial wall temperature differential (ΔT_w) shown in Figure 166 with the thermal cycles-to-failure prediction for the copper regeneratively-cooled thrust chamber, shown in Figure 159. Figure 166 also shows the expected coolant pressure drop and temperature rise between the fuel pump and chamber discharge manifolds, as a function of the throat wall barrier mixture ratio. Of particular interest is the fact that barrier mixtures near 5.0 cause a loss in thrust chamber life. This effect occurs because when changing mixture ratio from 6.0 to 5.0, the gas-side film coefficient increases more rapidly than the combustion temperature decreases. This is caused because of the good heat transfer properties of hydrogen, and increases the heat flux to the chamber. When the mixture ratio has fallen below 3.5, the flame temperature is sufficiently reduced that an increase in wall life is effected. In order that the chamber have a minimum low cycle fatigue safety of four, the design value of thermal cycles-to-failure (N_F) must be 1200, if the chamber is to be replaced at each engine overhaul period. Reference to Figure 165 shows that when a coolant channel Mach number of 0.45 is used at the chamber throat, a throat wall barrier mixture ratio of 3.2 is required. The operating conditions, as noted, are for 25,000 lbf thrust, engine mixture ratio of 6.0, and chamber pressure equals 1800 psia. In order that adequate supplementary cooling exists for a short ways downstream of the throat a barrier mixture ratio of 3.0 was selected for the engine. As the barrier mixture ratio increases aft of the throat the local pressure and heat fluxes fall rapidly, because of gas expansion, so that the local chamber life can be easily made greater than that at the throat by proper regenerative chamber wall design. Upstream of the throat, of course, the local wall barrier mixture ratio varies between 0.85 and 3.0, so that the local heat flux to the entire chamber is greatly reduced, and the local cooling requirements diminished. The use of barrier coolant, therefore, reduces the coolant pressure drop in the thrust chamber. This reduces the fuel pump discharge pressure and increases the engine controllability by increasing the turbopump power margin.

The use of barrier cooling has other beneficial [illegible] as well. It assures chamber compatibility, and strengthens the [illegible] of uniform circumferential thrust chamber wall conditions. It [illegible] reduce engine development costs significantly, because the

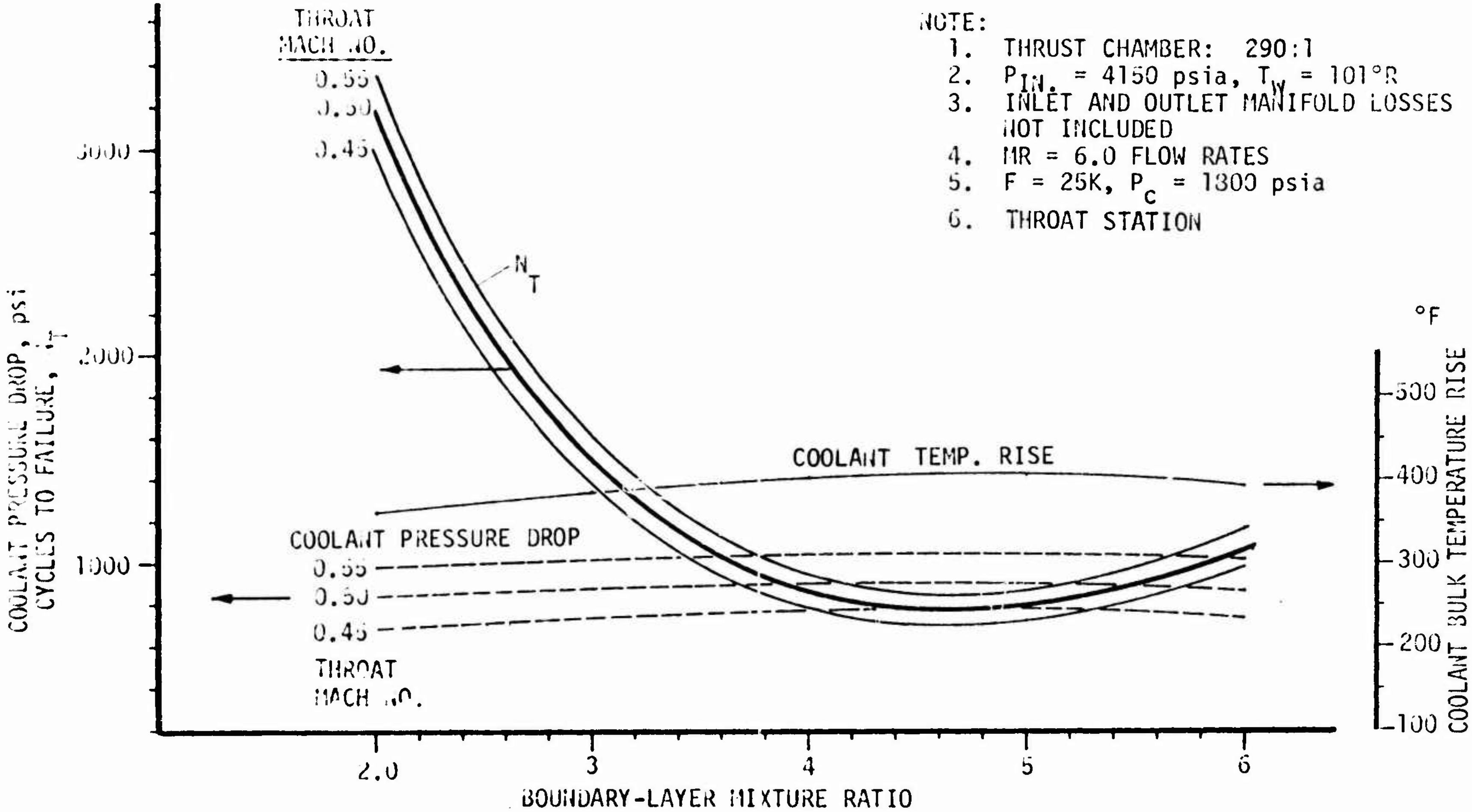


Figure 165. OOS Copper Nozzle Boundary Layer Mixture Ratio vs Life Cycle and Coolant Pressure Drop

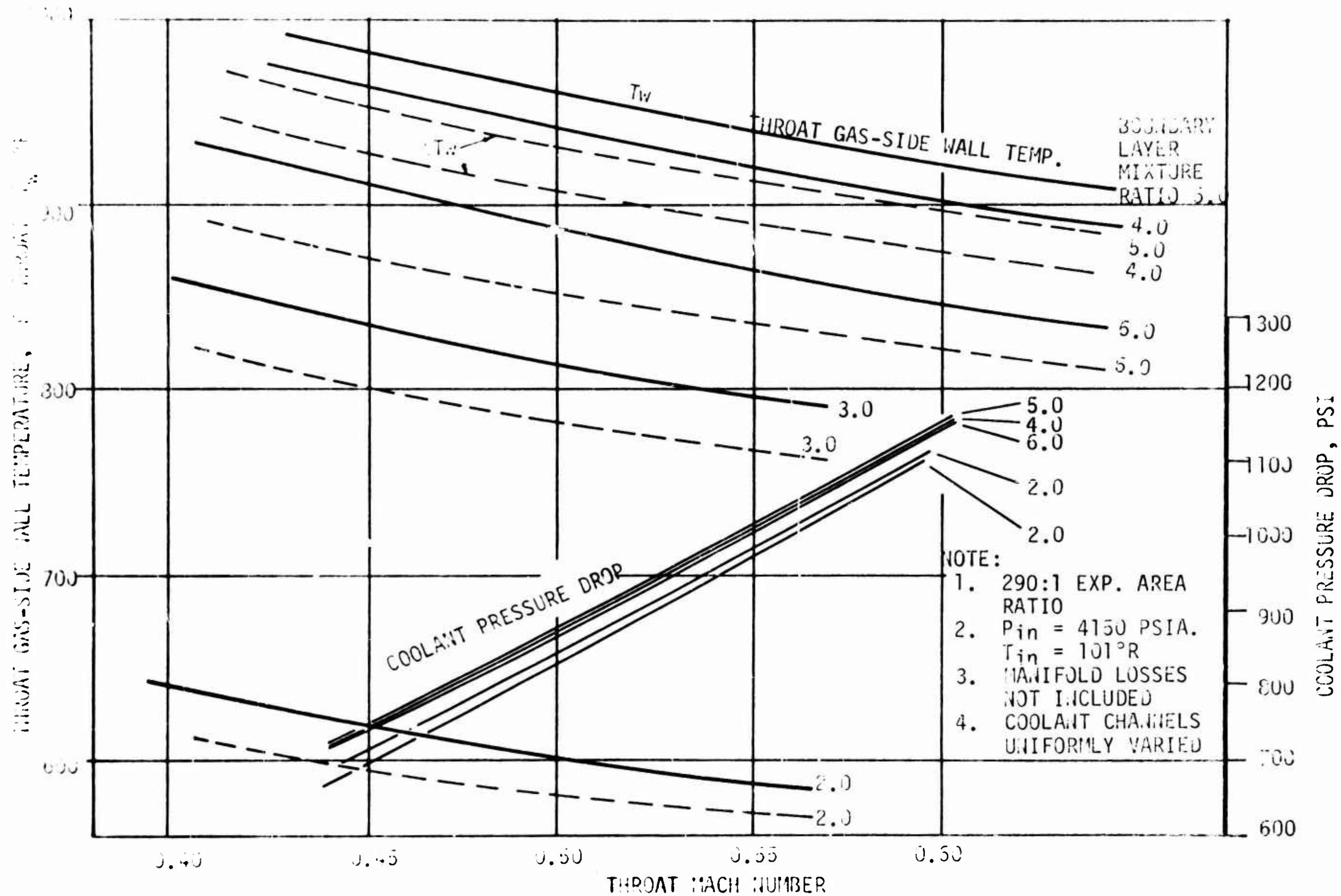


Figure 166. OOS Copper Nozzle Boundry Layer Mixture Ratio Study

chamber cooling margin may be easily altered by re-orificing the FRHG distribution plate located near the injector/hot gas inlet manifold bolted joint.

The expected engine performance loss caused by the use of the barrier cooling is 1.7 sec, or 0.36%. This is equivalent to a change in payload of 267 lbs; or 1.3% of a 20,000 lb nominal payload. The calculated MRD performance loss caused by the use of a throat barrier mixture ratio of 3.0 is 2.3 sec. However, if the barrier were not utilized the chamber life at 1800 psia operation would be only 1050 thermal cycles, and the N_F margin would be 3.5, or only 87.5% of the desired value of 4.0. In order to meet the N_F margin of 4.0 without the use of barrier cooling, the chamber pressure would need to be lowered to 1700 psia. The engine will not power balance at 1800 psia chamber pressure with the increased regenerative cooling pressure drop needed to obtain a chamber life margin of 4.0 at 1800 psia operating conditions without barrier cooling.

This reduction in chamber pressure would cause a reduction in nozzle area ratio from 290 to 270. The loss of available nozzle area ratio caused by the throat area increase of 6 percent is required to maintain thrust at 25K lbf, the nozzle length limit being unchanged. A reduction in specific impulse of 0.6 sec is associated with the nozzle expansion ratio reduction. Therefore, the net performance loss caused by the use of barrier cooling is only 1.7 sec. This tradeoff data as well as data for other conditions of engine life, performance, and cooling modes is given in Table XXXIV. The selected 600 cycle engine configuration operates at 1400 psia chamber pressure, uses a barrier mixture ratio of 3.0, and delivers 2.3 sec less performance than the selected 300 life cycle engine. The "60" life cycle engine uses no barrier coolant, operates at 1900 psia chamber pressure, and has a performance level 2.5 sec greater than the selected 300 life cycle engine. The "60" life cycle engine actually will deliver approximately 120 life cycles with an N_F margin of 4.0. This occurs because the gas-side wall temperature limit of 1000°F is reached, although the resulting life calculation at these conditions is 120 cycles. The use of higher wall temperature than 1000 is deemed unwise because of the lack of accurate material data at those elevated temperatures, and the indication by available data that strength, modulus, creep, and fatigue properties may rapidly deteriorate.

Table XXXV lists the effect of engine design mixture ratio on engine performance. The data shows that a slight performance increase of 1.1 sec is obtained by a shift to an MRE of 5.0. This is in spite of a loss of nozzle area ratio from 290 to 240, caused by the gamma induced nozzle contour change effect in a length-limited envelope. Conversely, operation at an MRE of 7.0 produces a significant performance loss of approximately 8.0 sec in spite of an increase in nozzle area ratio to 320:1. This assumes that the MRE = 5.0 and MRE = 7.0 engines both use a barrier mixture ratio of 3.0 at the throat and 1800 psia chamber pressure so that chamber wall gas-side conditions are similar. Greater barrier coolant injection flowrates are required with the MRE = 7 (Figure 165) engine, than for the MRE = 6.0 engine. The corresponding MRD loss is higher. Conversely, the barrier coolant injection flowrate and MRD performance loss of the MRE = 5.0

TABLE XXXIV

OOS ENGINE DESIGN ALTERNATES

ITEM NO.	DESCRIPTION	COOLING MODES NOZZLE	COOLING MODES CHAMBER	CHAMBER LCF SAFETY FACTOR	CHAMBER PRESSURE, PSIA	BARRIER M.R. AT THROAT WALL	CHAMBER MATERIAL	NOZZLE MATERIAL	NOZZLE AREA RATIO	CHANGE IN ENGINE SPECIFIC IMPULSE, SEC	PROBLEM
1	BASELINE DESIGN	REGEN ONLY	REGEN	3.5	1800	6.0	Zr-Cu	22-13-5	290	0.0 (NOMINAL)	LOW LCF
2	SKIRT DESIGN ALTERNATE	REGEN, TRANSPIRE, RADIATION	REGEN	4.0	1800	6.0	Zr-Cu	AGCarb-101	290	-<1.0	ADVANCED NOZZLE TECHNOLOGY
3	BARRIER COOLING ALTERNATE	REGEN ONLY	REGEN PLUS BARRIER	4.0	1800	3.0	Zr-Cu	22-13-5	290	-2.3	PERFORMANCE LOSS
4	LOW P_c ALTERNATE	REGEN ONLY	REGEN	4.0	1700	6.0	Zr-Cu	22-13-5	270	-0.6	WEIGHT INCREASE + PERFORMANCE LOSS
5	MATERIAL ALTERNATE	REGEN ONLY	REGEN	>4.0	1800	6.0	NARLOY Z	22-13-5	290	0.0	MATERIAL PROPERTIES UNCERTAIN

TABLE XXXV

OOS ENGINE PERFORMANCE VS. DESIGN MIXTURE RATIO

	P_c	R_t	MR_B	Life, Cycles	L_N, in.	ΔI_s, Sec Barrier	D_o, In.	ϵ_o	ΔI_s NOZ (MRE=6.0)	(Not Cooled MR = 6.0) ΔI_s Total
ENGINE MR = 5.0	1800	1.532	3.0	300	68.5	-1.3	47.5	240	+0.1	-1.2
ENGINE MR = 6.0	1800	1.51	3.0	300	68.5	-2.3	51.3	290	0 (Baseline,	-2.3
ENGINE MR = 7.0	1800	1.495	3.0	300	68.5	-5.0	53.5	320	-5.2	-10.2

engine is less than that of the MRE = 6.0 engine. The MRD performance loss is very sensitive to engine mixture ratio: this effect will be discussed below.

That barrier cooling is definitely required by both MRE = 5.0 and 7.0 design engines is clearly indicated by the data presented in Figure 163. This figure shows the results of an analysis done to determine how the throat life of a non-barrier cooled thrust chamber is affected by design engine mixture ratio. For a given coolant Mach number at the throat plane, the chamber life maximizes at MRE = 6.0. As shown in Figure 159, the chamber life is decreased by an increase in either gas-side wall temperature or wall radial ΔT, or both. Operation at MRE of 7.0 causes an increase in chamber gas-side wall temperature. The smaller coolant flowrate available results in the presence of warmer coolant in the cooling channels, and, hence the warmer gas-side temperature. Since the chamber wall backside temperature is also increased, the ΔT_w remains relatively unchanged. In the case of the MRE of 5.0 operation, the increased coolant flow causes a decrease in coolant temperature and backside wall temperature. However, the chamber gas-side wall coefficient increases so sharply with the additional hydrogen in the chamber gases, that the gas-side wall temperature is increased. Therefore, the T_{wg} and ΔT_w are both increased simultaneously, and the life of a non-barrier cooled thrust chamber falls rapidly.

Figure 167 shows representative specific impulse values versus mixture ratio with a constant nozzle expansion ratio. For chamber pressures near 1800 psia, the optimum performance occurs at a mixture ratio just below 6.0. When barrier coolant is injected separately from the balance of the propellant, the mixture ratio of the balance of the propellant is affected. Figure 168 is a plot of the injector "core" mixture ratio versus the boundary mixture ratio and the boundary flowrate, as percent of the total propellant flowrate, for MRE = 6.0. The performance loss is computed as a mass weighted average of the performances of the core and the barrier. The resulting performance losses are shown in Figure 169. When the engine mixture ratio is 7.0 rather than 6.0, the barrier cooling performance loss is greatly increased. First the core mixture ratio is shifted upwards from 7.0, in a region where performance losses increase rapidly with mixture ratio. Secondly, a greater amount of barrier coolant is needed to obtain the same throat barrier mixture ratio, and this shifts the core MR farther from the optimum of 5.8. Conversely, the MRD loss due to barrier coolant injection with MRE of 5.0 is quite small. The core mixture ratio is shifted toward optimum performance, and, also, the amount of barrier coolant flow required is reduced.

(3) Regenerative Thrust Chamber Cooling

The above remarks have presupposed knowledge of the regenerative cooling portion of the thrust chamber. The following is a discussion of the regenerative cooling features of the chamber and its interaction with the engine.

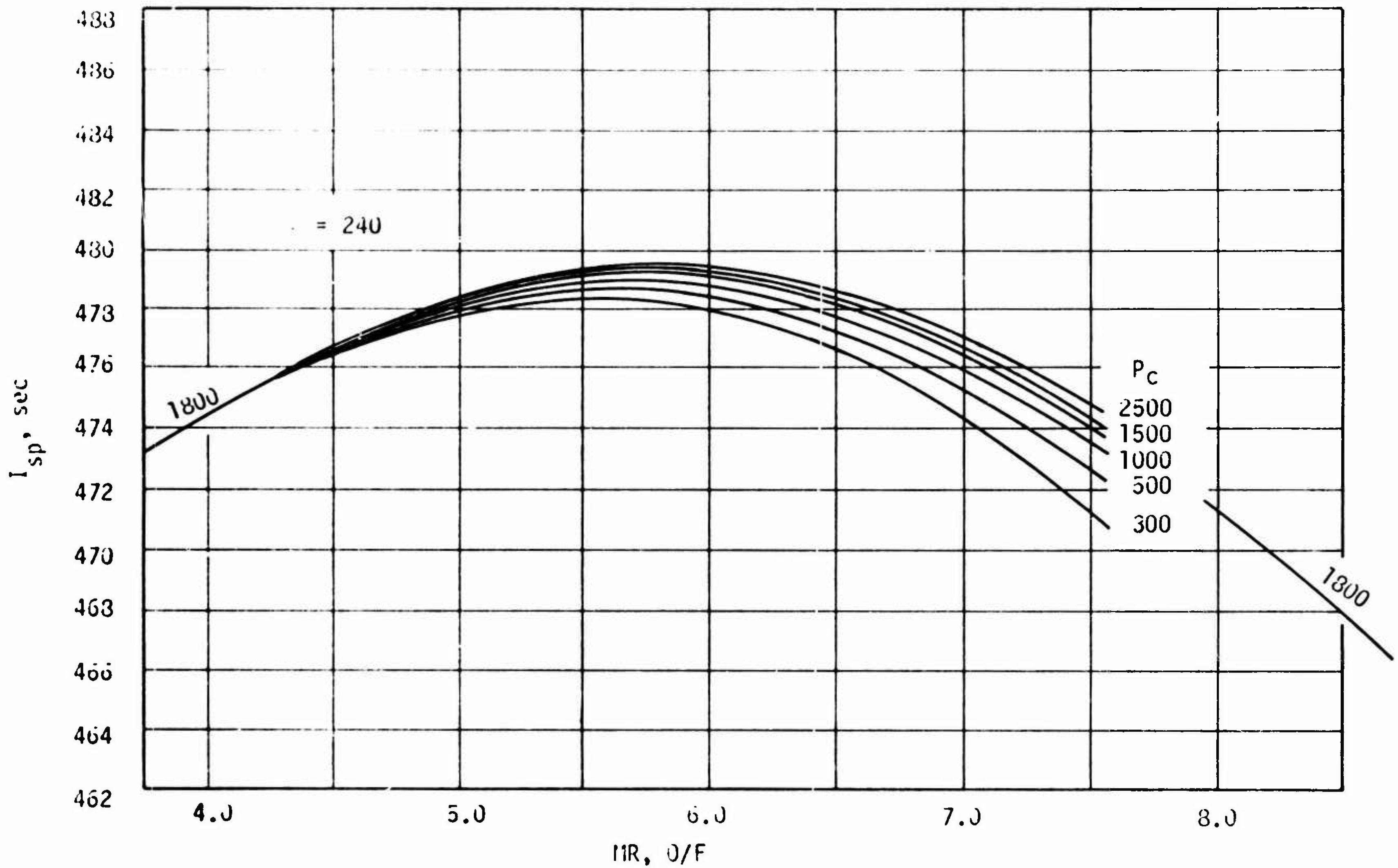


Figure 167. Specific Impulse vs Mixture Ratio at Constant Area Ratio, 25K

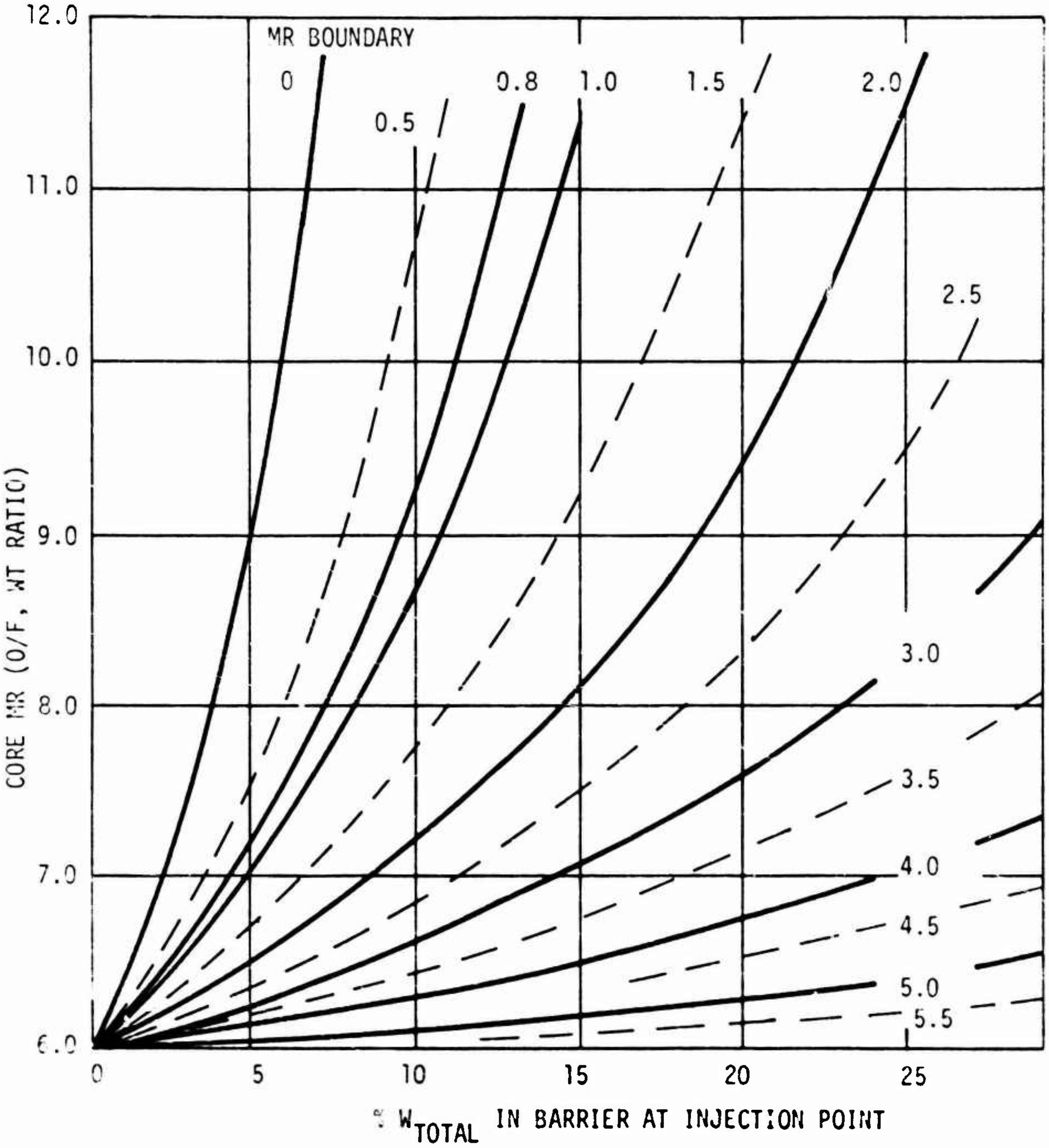


Figure 168. OOS Barrier Cooling - Core Mixture Ratio for MR_E = 6.0

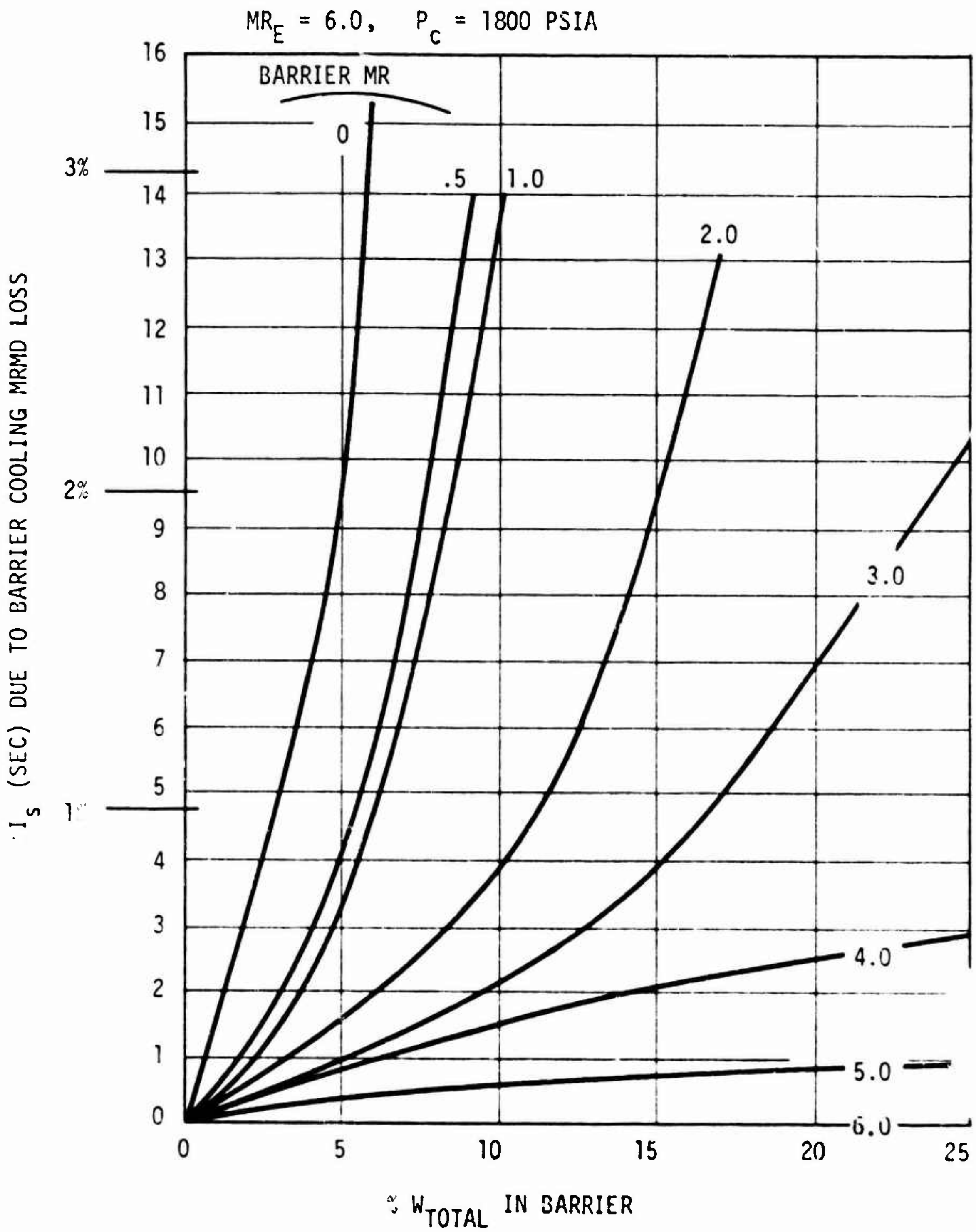


Figure 169. OOS Barric· ·ooling Loss Estimate

For regenerative cooling to be feasible at the very high heat fluxes of the OOS engine, the chamber must be designed to maximize the two-dimensional conduction capability of the inner wall. Copper and some of its alloys are the only practical materials with sufficient thermal conductivity (together with the other required properties) to transfer the very high heat load to the hydrogen coolant. The selection of zirconium copper material was based upon its superior low cycle fatigue characteristics relative to all other alloys considered and tested at ALRC.

In order to maximize the two-dimensional conduction capability of the inner wall, the chamber must be designed to maximize the effective coolant side surface area. Significant enhancement of the cooling capability of the wall in this way depends upon a fin effect, and is maximized by employing the largest number of very small cooling passages as is practical. The use of small passages allow more fins and also provide the necessary support for the necessarily thin gas-side wall. Two factors which are of major importance in maximizing the wall fin effects are geometry and wall material. The geometric influence was investigated via a two-dimensional conduction network computer program (SINDA 3G) wherein typical throat gas-side conditions were imposed upon various coolant channel geometries. The gas-side wall thickness of 0.030-in. was held constant for this study. This was considered the minimum practical wall thickness from fabrication considerations. Figure 170 demonstrates the influence of channel geometry upon the two-dimensional conduction effects of the wall by showing the effect of a land width to channel width ratio on gas-side wall temperatures at the throat location. It can be seen that the optimum channel geometry (L/W 1.0) results in a gas-side wall temperature much lower than the one-dimensional conduction approach would show. The optimized wall temperature is much closer to that predicted by a one-dimensional fin analysis. This optimized geometry is a function of the applied heat flux and hence, varies along the chamber contour. At the injector end of the chamber where the non-barrier cooled heat flux is only 60% of that at the throat, the optimum land width to channel width ratio increases to about 2.0.

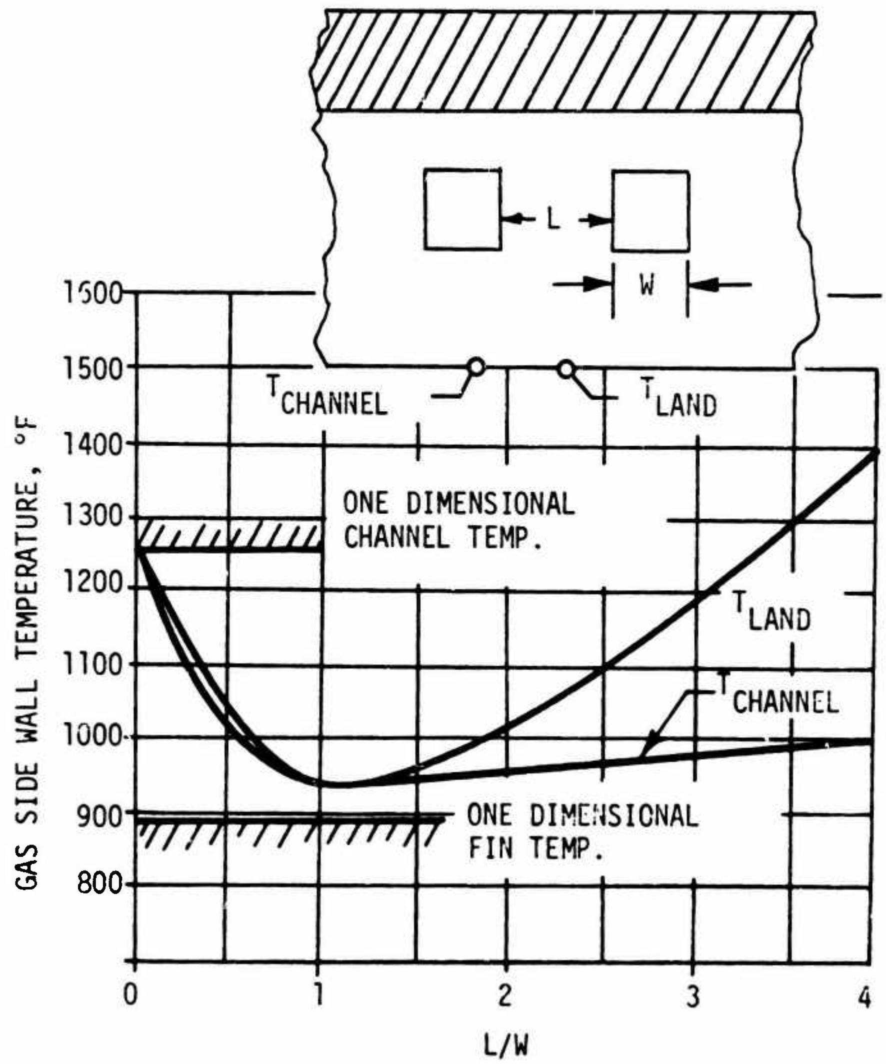


Figure 170. Influence of Channel Geometry on Chamber Cooling Characteristics at Throat Conditions

c. Nozzle Concept - Fixed and Retractable

The primary function of a rocket engine nozzle is to expand the gases generated in the thrust chamber to obtain a higher engine specific impulse than would otherwise be obtained. For this reason considerable increase in engine size, weight, and cost is willingly accepted by the addition of the nozzle to the aft end of the thrust chamber. The nozzle must also be cooled, and while the heat flux to the majority of the surface is low, the total cooled area is large, so that the total heat load is significant, and can affect the design of the entire engine.

Hydrogen/oxygen engines which operate in a vacuum environment can theoretically use very large expansion area ratio nozzles for the purpose of extracting the maximum performance from the thrust chamber gases. Nozzle weight, cooled area, and nozzle center of mass distance from the engine center of mass all increase rapidly with area ratio, however. It is possible to determine the optimum nozzle expansion area ratio if the proper trade-off-parameters are known. The resultant specific impulse performance level is usually less than the maximum obtainable, but the system performance is maximized. Where severe engine length limit exists, the "optimum" nozzle area ratio is the maximum obtainable and can be determined by length alone. In this case, it is necessary to utilize an optimum nozzle contour to assure the attainment of the maximum nozzle efficiency vs nozzle length. Further, the remainder of the engine length must be minimized to allow the use of the longest possible nozzle. In cases where the lack of available nozzle length greatly reduces engine performance, it may be advantageous to stow a portion of it around the engine, and extend it into position when ready for use. The packaged engine dimensions are, therefore, small, and the nozzle area ratio and performance level high. Unfortunately, this nozzle design concept increases the nozzle and engine weight, complicates the nozzle cooling, and adversely impacts the engine cost and reliability factors.

(1) Retractable Nozzle

Data generated at ALRC early in the OOS contract showed that the optimum weight/performance engine tradeoff occurred with the use of an extendable-retractable nozzle, whose movable section had an inlet area ratio of 125:1 and an overall area ratio of 450:1. The effective use of such a large area ratio nozzle was made possible by the selection of a fibrous graphite wall material, AGCarb-101 for the movable section. The AGCarb-101 properties of high service temperature capability, high strength at elevated temperature, low density, etc. caused the specific wall weight to be less than one-half of that of a similar nozzle made of Columbium. The movable nozzle was cooled by radiation, so that no coolant lines or fittings were required to be connected to the movable components. The fixed portion of the nozzle, extending from area ratio 6 to 125 was regeneratively-cooled by hydrogen and of tubular construction. It was determined that coolant [illegible] along the inner wall of the movable nozzle was required to cool [illegible] movable nozzle interface metallic joint, as the adiabatic wall

temperature of the AGCarb-101 nozzle at full thrust operation was 3000°F. The coolant used was hydrogen, and was tapped off the tubular nozzle at the coolant turn-around manifold, and injected along the graphite wall supersonically through platelet-type two-dimensional nozzles, as shown in Figure 171. The coolant injection performance loss was estimated and reported in the 90-day data dump, to be approximately 0.9 sec of specific impulse.

(2) Fixed Nozzle

As the engine studies continued, it became increasingly obvious that the slight increase in overall vehicle payload provided by the use of the extendable-retractable nozzle would actually be only about 100 lbs, or 0.5% of a 20,000 lb payload. Considering all cost and reliability factors, it was decided that the preferable OOS 25K engine should use a fixed nozzle. The fixed nozzle area ratio which fits the envelope is 290:1, when used in conjunction with the 1800 psia thrust chamber assembly and minimum length Rao optimum nozzle contour. Accordingly, the extension-retraction mechanism was removed from the engine design and the AGCarb-101 nozzle extension was reduced in length to provide 290:1 area ratio. The tubular regeneratively-cooled nozzle/graphite skirt joint was redesigned as shown in Figure 172, to provide a bolted joint and the same coolant injector as was previously used. The AGCarb-101 skirt extension was retained in the fixed nozzle design, rather than switching to an all regeneratively-cooled nozzle for several reasons. First, the specific wall weight of a regeneratively-cooled nozzle is 2.5 times that of a similar graphite nozzle. Therefore, in spite of the need for a large diameter mounting joint, the graphite/regenerative design was lighter than the all regenerative design by 30 lbs. This very nearly offset the expected graphite extension coolant injection performance loss of 0.9 sec I_s with respect to vehicle payload calculations. In addition, the graphite extension, being of large area and radiation-cooled, was desirable because it reduced the heat load to the coolant considerably, compared to the all regeneratively fuel cooled design.

A weight and performance comparison between the all regeneratively-cooled nozzle and the radiation-cooled nozzle extension with the above cooling requirement revealed no payload advantage for the considered area ratio of ε = 290:1. It was decided to use the all regeneratively-cooled nozzle in order to eliminate all engine heat radiation and thus obtain engine application feasibility and a thrust chamber configuration of proven technology.

Fixed Nozzle Concept Description

The selected nozzle design is shown in Figure 156 attached to the thrust chamber. The basic nozzle design specifications are given in Table XXXVI. The nozzle coolant inlet manifold is a tapered torus located at the small end of the tube bundle. It receives hydrogen coolant directly from the fuel pump. The hydrogen enters alternate "down" tubes through the copper ring joint at the nozzle forward end shown in the detail

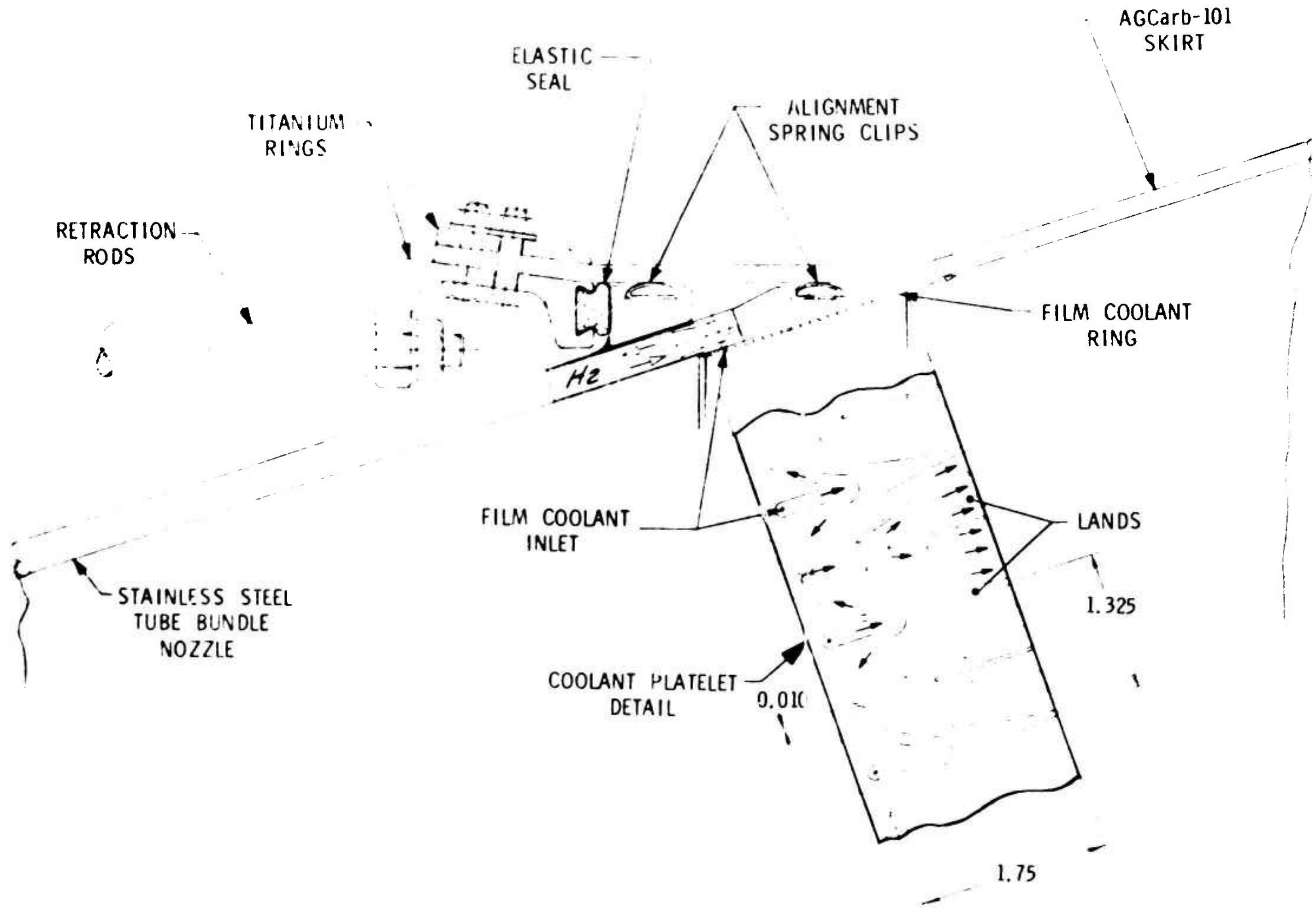


Figure 171. OOS Extendible Skirt Joint Detail

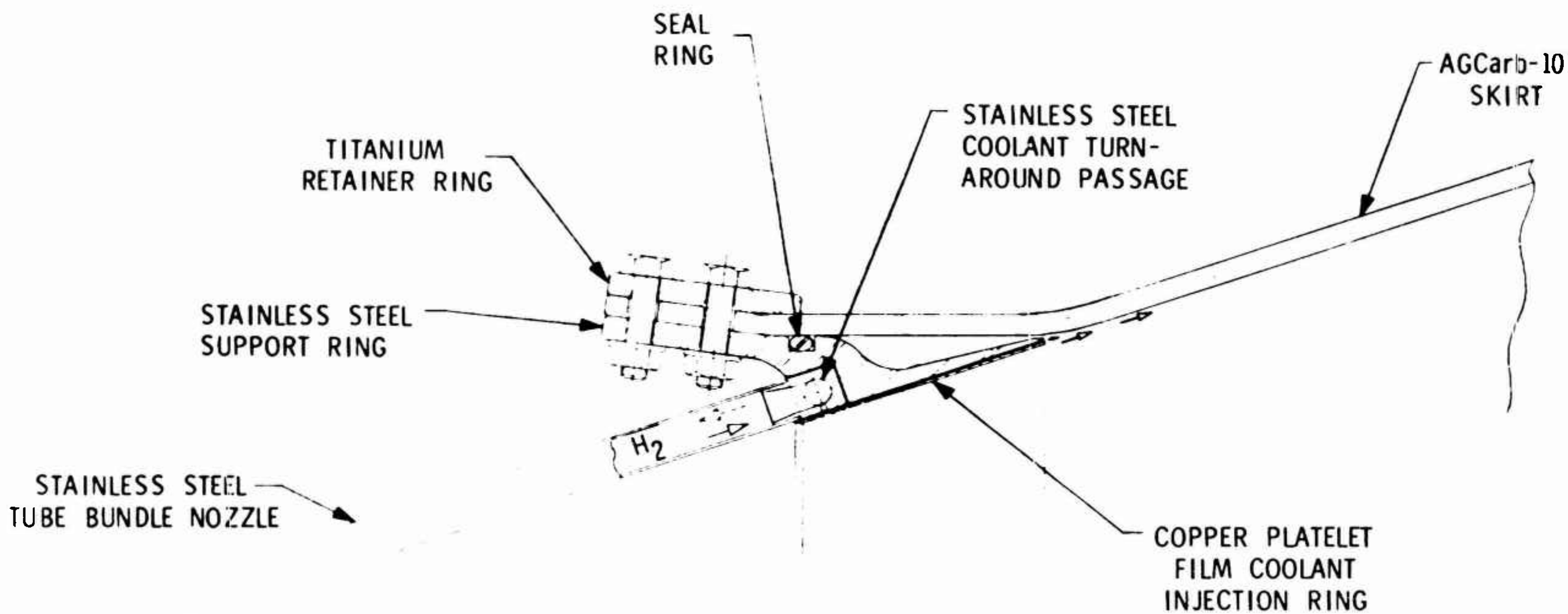


Figure 172. OOS Fixed Skirt Joint Detail

TABLE XXXVI

NOZZLE BASIC DESIGN SPECIFICATIONS

INLET AREA RATIO	6:1	COOLANT SCHEME	TWO-PASS HYDROGEN REGEN.
EXIT AREA RATIO	290:1	BIFURCATION JOINT TYPE	CIRCULAR TUBE SURROUNDING BACK-TO-BACK "D" TUBES, BRAZED
EXIT DIAMETER, IN.	51.4	TURNAROUND MANIFOLD TYPE	"U" TUBULAR AT ϵ = 290, BRAZED
LENGTH, IN.	65.2	NUMBER OF TUBES:	TOTAL SEGMENTS = 1330
CONTOUR	MINIMUM LENGTH RAO OPTIMUM	ϵ = 6 TO ϵ = 30	190
CONSTRUCTION TYPE	ROUND TUBULAR-FURNACE BRAZED	ϵ = 30 TO ϵ = 125	380
TUBE WALL THICKNESS, IN.	0.015, CONSTANT	ϵ = 125 TO ϵ = 290	760
TUBE MATERIAL	ARMCO 22-13-5, TAPERED TUBES	NUMBER OF STIFFENING RINGS	6
NUMBER OF TUBE BIFURCATION PLANES	2	ATTACHMENT OF THRUST CHAMBER	REBRAZABLE JOINT
LOCATION OF BIFURCATIONS	ϵ = 30 AND ϵ = 125	NOZZLE WEIGHT, lb	107.4

view of Figure 173. As the hydrogen flows aft it splits into two adjacent tubes at area ratio = 30 and into four adjacent "down" tubes at area ratio = 125, arriving at the nozzle exit plane at very low velocity and high pressure. Coolant turn around is accomplished through concentric pairs of swaged ended, round cross section, "U" tubes, which turn coolant into first and second adjacent tubes on each side of the four adjacent "down" tubes. Therefore, coolant returns in four adjacent "up" tubes, merging into two at area ratio 125 and one tube at area ratio 30. The heated coolant arrives at the aforementioned copper ring joint in alternate "up" tubes, which dump into a small cross section, constant area torus plenum, which in turn feeds the coolant channels of the copper thrust chamber inner wall. Nozzle tube cross section is round at every axial station (except at bifurcations), such that tube diameter varies directly with the nozzle diameter between bifurcation joints. At the joints, the two smaller tubes are locally formed into "D" shapes, such that when assembled their flat sides fit back-to-back, and the two tubes together, form a single round tube with a double walled diametral baffle. This subassembly slips inside of the corresponding larger, single downstream tube. This joint design features low weight, rapid assembly, minimum number of parts, and can be sealed completely in a single furnace braze heat.

The tube material is ARMCO 22-13-5, and was chosen for its combination of properties including hydrogen embrittlement resistance, high strength, good low cycle fatigue properties, brazability and weldability. Higher tube strength than that afforded by ARMCO 22-13-5 was not needed, because the minimum wall thickness considered practical for a reusable, maintainable nozzle of 0.015-in. adequately contains the coolant pressure. A complete structural and low cycle fatigue analysis of the nozzle design is given in Appendix C. The nozzle fatigue life of 2550 cycles with N_F safety factor of 4.0 is limited by conditions at the tube crown at the nozzle inlet (ϵ = 6.0). Figure 174 shows the tube life as a function of axial distance from the engine throat. The life is such that the nozzle is a reusable component at each engine overhaul. The copper thrust chamber, however, may not be, and may need to be replaced as often as each overhaul if a N_F safety factor of 4.0 is to be maintained.

A braze joint was utilized at the nozzle/chamber interface for several reasons. First, this was the lightest weight joint design. Secondly, seal reliability problems caused by distortion of hardware, etc., are eliminated. Thirdly, the number of thrust chamber coolant channels and nozzle tubes differ so that a common interstitial coolant plenum is required in any case. Fourthly, a gradual inner wall contour change from smooth to convoluted is desired at the joint. The copper ring portion of the nozzle provides the surface which may be electrical discharge machined (EDM) to the desired internal transition geometry. As shown in Figure 173, the braze joint is designed to be well-cooled because of the high local coolant velocity caused by the small local plenum size. The joint is a copper-to-copper joint, so local thermal gradients, and low cycle fatigue cracking problems are minimized to meet the engine requirements. The joint is designed also with only one diametral and one planar surface requiring close tolerance machining. The outer diametral joint is not required for strength,

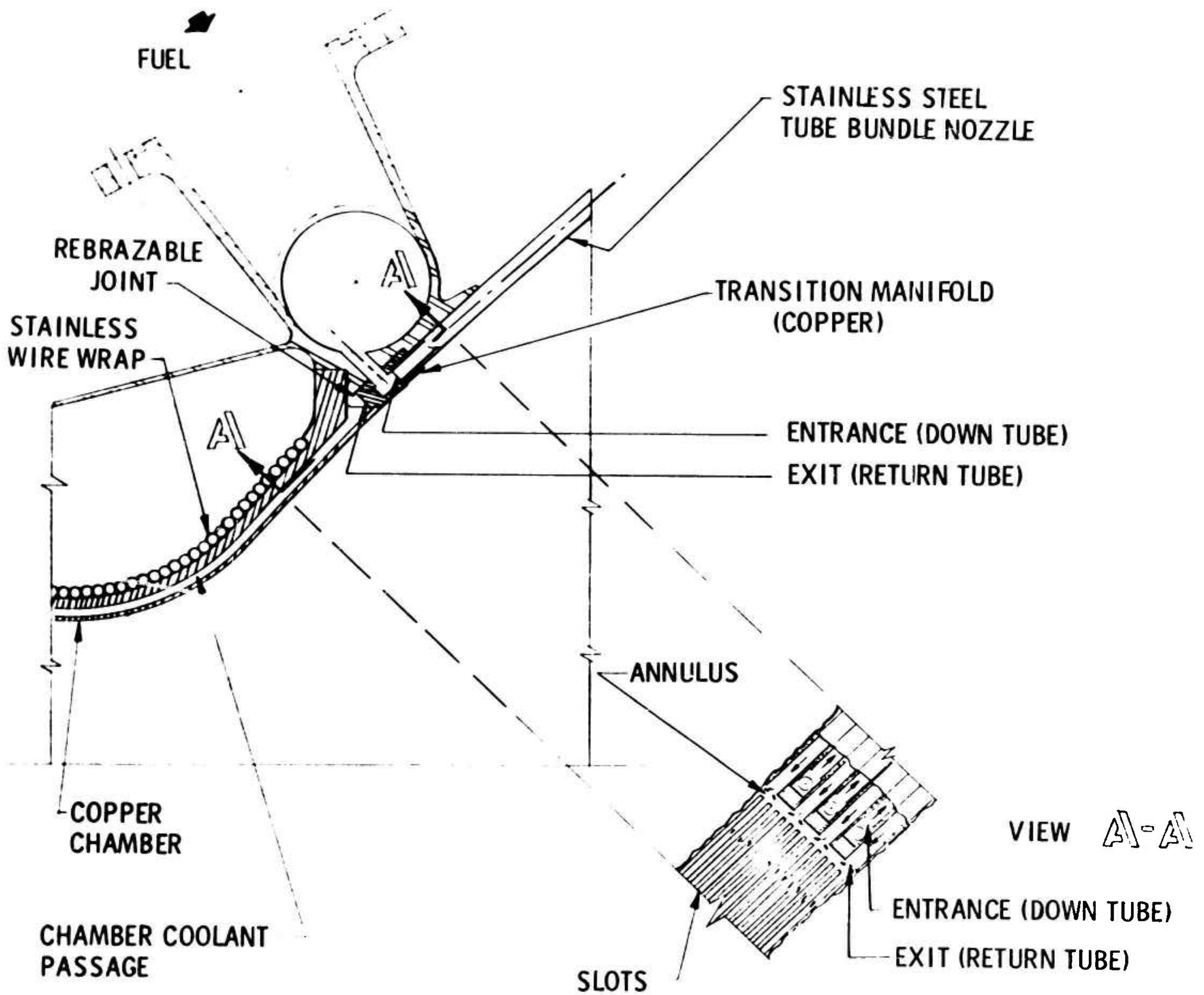


Figure 173. OOS Thrust Chamber and Nozzle Assembly

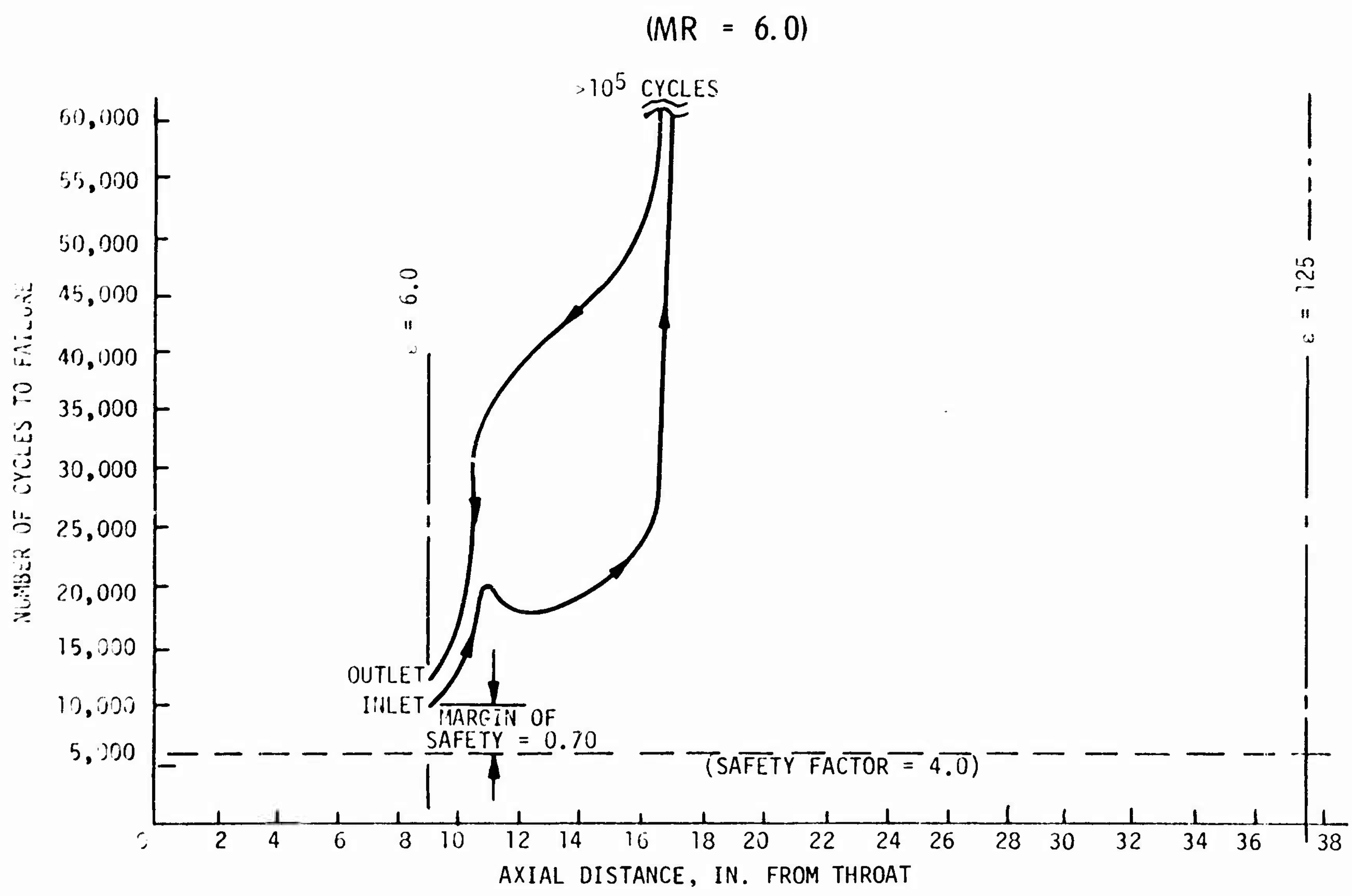


Figure 174. OOS Regenerative Nozzle Tubes LCF Life Predictions

and is intended to feature a large gap to be filled with braze alloy for sealing redundancy only. The geometry of the joint allows the gas-side braze joint to have a significant width in spite of the thin wall thickness normal to the chamber wall just above the joint. Therefore, the design is intended to allow subsequent, new thrust chambers to be furnace-brazed with a low temperature alloy to a used nozzle, after appropriate removal of the old chamber.

The thermal analysis of the nozzle is presented in Appendix B. Nozzle weight is listed in Table XXVI as 107.4 lbm along with the other major combustion components.

<u>Dump Cooled Nozzle Extension</u>

A tubular dump cooled nozzle was examined for application to the subject engine. With a dump cooled design, hydrogen is bled from the thrust chamber inlet manifold through single pass tubes and expanded through small nozzles at the exit end of each tube. Because the tube internal pressure would be low, it was expected that a light weight nozzle would result. It was also expected that the hydrogen coolant would produce very high performance. In addition, the coolant pressure loss otherwise suffered in a regeneratively-cooled nozzle would not be present.

Comparison with the regeneratively-cooled design showed that: (1). There was very little weight difference between the regeneratively- and dump-cooled nozzle because minimum handlable gage tube walls are used in either design; (2). An overall performance loss is suffered with the dump-cooled design because of the shift in thrust chamber mixture ratio toward oxidizer-rich; (3). The coolant pressure drop in the regeneratively-cooled nozzle design is very small compared to the pressure loss in the thrust chamber coolant passages. The dump cooled nozzle design was not, therefore, selected because of the performance loss associated with it. Additionally, it was not desired to cool the thrust chamber with liquid hydrogen. Rather, it is desirable to have a certain amount of pre-heat supplied to the coolant prior to introduction to the high heat flux, highly strained region.

d. Preburner Assembly

The primary function of the preburner in a staged-combustion engine is to produce and deliver a homogeneous mixture of fuel-rich hot gas (FRHG) to the turbine inlet manifold. The preburner receives heated hydrogen from the thrust chamber regenerative coolant circuit outlet and liquid oxygen from the second-stage oxidizer pump discharge through a control valve. The preburner must inject the propellants in such a manner that they may be ignited and efficiently combusted in a smooth, repeatable manner without allowing damage to the injector, chamber, or turbine assemblies over a period as long as 50 hours, including 1500 thermal cycles.

In the selected design, the preburner chamber has several additional functions. It must contain the high pressure FRHG, direct it to the turbine inlet manifold, and form a structural, leak-free connection between the injector and turbine manifold. It is constructed such that it contains an acoustic damper near the injector face, in order to control longitudinal high frequency combustion instability. The preburner chamber also acts as a FRHG/oxidizer heat exchanger for the following reasons:

(1) Heat the liquid oxygen to supercritical temperatures at all engine thrust levels.

(2) Cool the preburner chamber wall.

(3) Provide an autogenous tank pressurant source for the oxidizer tank without the use of a separate heat exchanger for that specific purpose.

The advantages of heating the oxidizer prior to injection are similar to those given in Section III.B.2.a. for the main injector oxidizer heating. The difference is that the oxidizer is heated in chamber channels rather than in the injector because the inlet fuel temperature is low at 450°R, whereas the main injector FRHG inlet temperature is 1700°R.

The preheating of the oxidizer disallows liquid or two-phase injection through the relatively large, easily fabricated, preburner injection orifices sized for gaseous injection. Therefore, serious time variant as well as space variant mixture ratio shifts (upward) are avoided, and primary turbine blade protection is assured. During preburner starting, the oxidizer is gassed off by the unchilled feedline, manifolding, coolant passages, and nickel platelet type injector. As soon as the preburner is ignited, the oxidizer heating source is available, and oxygen heating assured. At supercritical temperatures, the oxidizer density varies rapidly with pressure. A very wide excursion in oxidizer preburner flowrate of almost 15:1 can be required by the 5:1 engine throttling requirement and off mixture ratio engine capability of from 5.5 to 6.5:1. The oxidizer density change with preburner pressure caused by the supercritical temperature of the

oxidizer allows the use of a single oxidizer injection circuit, and single circuit, remote, liquid oxidizer control valve, without resorting to high inlet oxidizer pressures at full thrust operation. This effect is shown in Table XXXVII, which illustrates that the change in oxidizer injection stiffness from full thrust to 20% thrust is much less than the factor of 15 which would otherwise be expected with liquid oxidizer injection.

Preburner Operational Analysis

The data in Table XXXVII was generated assuming that the preburner fuel bypass valve remained closed during the entire thrust/MRE excursion. This is the worst case for preburner oxidizer heating, because under these conditions the preburner chamber oxidizer outlet temperature falls at the maximum rate with throttling, and this reduces the oxidizer stiffness at the maximum rate. The calculated drop in oxidizer temperature is illustrated in Figure 175, which shows oxidizer temperature, pressure and density as functions of engine thrust level and axial station in the chamber, from the inlet manifold. It is noteworthy that the design allows for significant superheat at minimum thrust, even under these most adverse conditions. The causes of the oxidizer temperature drop with throttling in the face of the decreased oxidizer preburner flowrate are:

(1) Decrease in preburner chamber gas mixture ratio (and temperature) caused by a rapid decrease in TPA power requirements.

(2) Decrease in chamber coolant passage gas-side heat transfer coefficient because of reduced chamber pressure.

Proper sequencing (opening with throttling) of the preburner fuel bypass valve, however, causes the preburner mixture ratio to fall less rapidly and boosts the oxidizer temperature. This increases the superheat margin and reduces the oxidizer density, thus increasing the throttled preburner oxidizer stiffness so that the stiffness remains nearly constant over the entire engine throttling range. This allows the use of the minimum possible oxidizer inlet pressure at full thrust conditions, which reduces pump power requirements, and pump and preburner weights. The required maximum oxidizer pump discharge pressure is less than 4700 psia with the selected engine system and preburner design. With liquid injection, it would be greater than 10,000 psia.

The preburner mixture ratio control provided by the fuel bypass valve reduces the preburner assembly weight and improves its reliability by maintaining the mixture ratio above the flammability limit at all engine thrust levels. Because of this, axial preburner main chamber ignition can be utilized, rather than a separate torch igniter. At operation below the flammability limit, the torch heat would need to be continuously supplied from the chamber wall across the injector face, the only place in

Table XXXVII. Preburner Operating Conditions

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
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[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.139	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	147	0.156	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.158	[illegible]	[illegible]	[illegible]
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[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	343	0.121	[illegible]	[illegible]	6.0
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.131	[illegible]	[illegible]	7.6
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	0.141	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	121	0.173	[illegible]	[illegible]	13.4
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	6.5
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	7.5
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	10.5
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	16.0
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

*DOES NOT INCLUDE VALVE (THROTTLEABLE) AND DISTRIBUTION SYSTEM

**INCLUDES GAS DISTRIBUTION PLATE

$A_{FJ\ TOT} = 1.04\ IN.^2$

$A_{OJ\ TOT} = 0.020\ IN.^2$

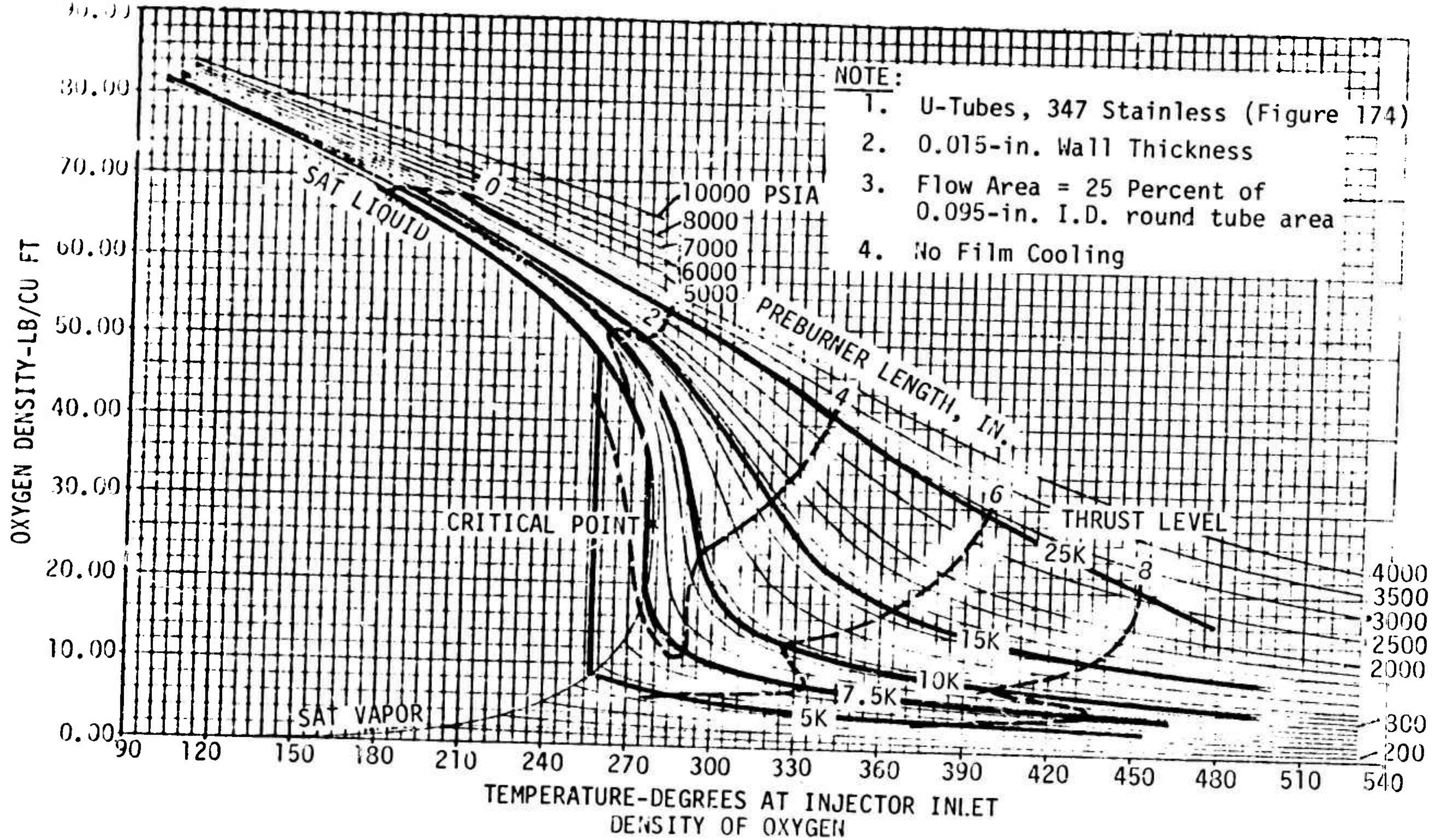


Figure 175. Effect of Preburner Length at Varying Thrust Levels

the chamber where ignitable, high local mixture ratios exist. In that case, the igniter would need to operate continuously to prevent preburner flameout during deep throttle operation.

Design Consideration

Heating the oxidizer prior to its introduction into the nickel - 200 injector has the additional advantage of eliminating the temperature difference between the two inlet propellants. Injector internal heat transfer and associated low cycle fatigue failure problems are thereby eliminated. Ni-200 is shown to be the required injector face material for use in high pressure O_2/H_2 preburners on the ALRC SSME program. Stainless injectors suffered from face erosion, while similar Ni-200 units did not, because of the difference in thermal conductivity between the two metals.

Regenerative cooling the preburner chamber wall has the advantage of reducing the chamber weight. The structural backside section of the wall, remaining cool, retains its mechanical properties, and allows the use of thin sections in spite of the high pressure, high temperature internal chamber conditions. With regenerative cooling, the wall induces plastic strain in the gas side of the coolant passages and limits the fatigue life of the chamber. The chamber life is sufficient to provide an N_F margin of greater than 4.0 for the entire engine life of 1500 thermal cycles, as shown in Figure 176. It should be noted that the use of regenerative cooling does not cause the LCF limit, it merely fixes the single location of the effect. An uncooled chamber would, if feasible, need to be attached to cooler structures, and the problem would be displaced to the fore and aft joints. Furthermore, the coolant-passage gas sides are quite flexible so that the LCF life i increased beyond that reasonably expected in a rigid joint The LCF life is a function of chamber length and is shown in Figure 176. Finally, the coolant can be efficiently utilized with the selected concept; cooled joints often require fluid flow paths which would not otherwise be required, and, hence, complicate the engine system.

The selected preburner design provides heated oxygen for autogenous oxygen tank pressurization at 400°R. The need for a separate heat exchanger is thereby eliminated. In its place, a pressure reducer is required, which can be as simple as a series of sonic orifices. The heat exchanger which would otherwise be required would probably receive its heat from warmed hydrogen (or possibly FRHG). In either case, a crack in the heat exchanger would allow mixing of oxidizer and fuel, which would eventually or immeidately ignite, producing a catastrophic failure. The use of oxidizer to regeneratively cool the fuel-rich preburner chamber is, therefore, no worse than the use of an autogenous oxidizer heat exchanger, from the failure criteria standpoint. The effect of combustion in the preburner chamber is minimized by the use of a fuel regeneratively-cooled inner wall extending three inches aft from the injector face. The majority of mixing

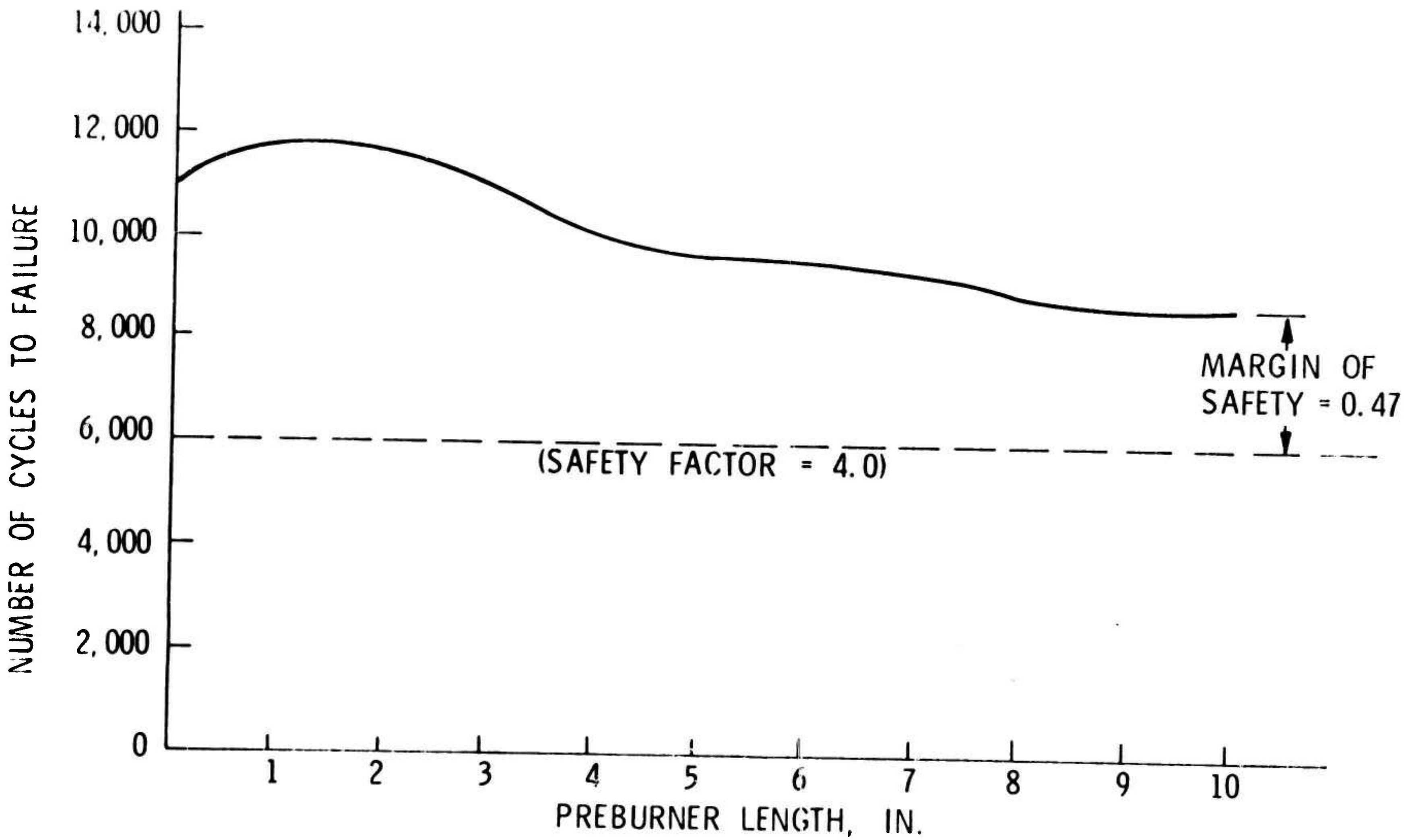


Figure 176. Preburner Low Cycle Fatigue Life at 25K Thrust

and energy release will occur in this region of the chamber.* The last six inches are required to allow secondary mixing to reduce the high local mixture ratio caused by the igniter oxidizer flow prior to entry into the turbine inlet mixing manifold. In addition, the majority of the heat transfer from the combusted gases to the oxidizer is completed in these six inches.

The forward portion of the preburner chamber, being double walled and separated by ring baffles, forms acoustic liner cavities. Local dimpling of the preburner fuel regenerative tubes provides the required orifices which connect the chamber to the resonators. The selected OOS preburner design is shown in Figure 177. Its basic design specifications are listed in Table XXXVIII.

The injector is a photoetched and brazed assembly of 0.010-in. thick Ni-200 platelets contained within an ARMCO 22-13-5 housing. A total of 5986 rectangular injection orifices 0.010 x 0.020-in. are provided in the three inch diameter face. The injector pattern shown in Figure 178 contains 1997 self-impinging fuel doublets and 996 similar oxidizer doublets. The doublets impinge at an angle of 60°. The fuel doublets impinge 0.030-in. from the injector face in order to shield it from the oxidizer. The very fine injector pattern was selected on the basis of SSME preburner tests at ALRC which verified that very uniform thermal profiles could be generated at low (preburner) mixture ratios with this type of injector. The injector feeds the central igniter circuit containing a spark electrode with oxygen, and the igniter fuel is provided at the injector face with the main propellant injection. Therefore, no preburner igniter valves are required.

The chamber design, shown in Figure 177 consists of ARMCO 22-13-5 hydrogen-cooled regenerative two-pass inner-wall near the injector face. The fuel inlet is through a tapering toroidal manifold surrounding the injector. A portion of the fuel is bled directly into the injector fuel inlet manifold and injected largely in the peripheral fuel orifices. The remainder passes through the tube bundle and is injected near the central igniter. The remainder of the chamber is cooled with oxygen in a single-pass, counter-flow manner, containing a tapered toroidal oxidizer inlet manifold and a small cross section collector manifold and feed passages to the oxidizer injector inlet manifold at the forward end of the injector periphery. The chamber construction, as shown in Figure 177, is an electron beam welded assembly of individually welded and flow tested ARMCO 22-13-5 "T tubes". The "tubes" are fabricated from formed open channels and stamped and machined closure strips. When assembled, the tubes provide high velocity, small hydraulic diameter coolant passages, ideal for heat exchange and a cooled, structural, sealed backside wall.

* Any tube failure in the hydrogen-cooled wall caused by local mixture ratio variations would not be catastrophic, but rather degrade the engine thrust level slightly.

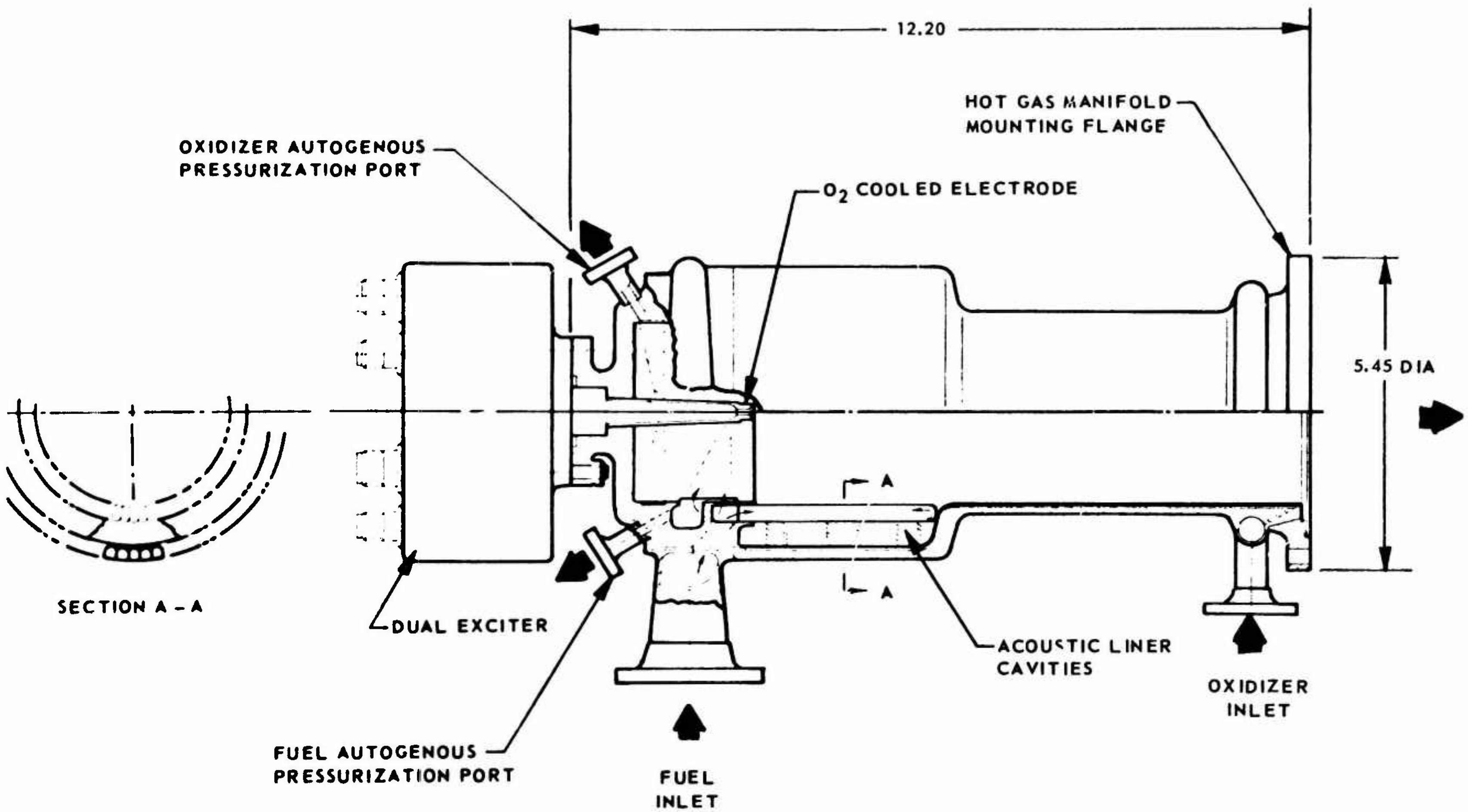


Figure 177. OOS Preburner Assembly

TABLE XXXVIII

PREBURNER BASIC DESIGN SPECIFICATIONS*

INJECTOR TYPE AND MATERIAL	BRAZED PLATELET, N_i-200	FUEL INJECTION VELOCITY, FT/SEC	1000
PROPELLANT INJECTION PHASE	GAS/GAS	OXIDIZER INJECTION VELOCITY, FT/SEC	125
INJECTOR PATTERN	LIKE-ON-LIKE DOUBLETS	FUEL INJECTION TEMPERATURE, °R	450
NUMBER OF ORIFICES		OXIDIZER INJECTION TEMPERATURE, °R	400
FUEL	3994 → 1997 DOUBLETS		
OXIDIZER	1992 → 996 DOUBLETS		
TOTAL =	5986 → 2993 DOUBLETS	TOTAL FUEL PRESSURE DROP, PSI	320
ORIFICE SHAPE AND SIZE	RECTANGULAR	TOTAL OXIDIZER PRESSURE DROP, PSI	600
FUEL, IN. x IN.	0.010 x 0.020		
OXIDIZER, IN. x IN.	0.010 x 0.020	CHAMBER TYPE	OXIDIZER REGEN. COOLED, WITH FUEL REGEN. COOLED LINER
DOUBLET IMPINGEMENT ANGLE			
FUEL AND OXIDIZER, DEGREES	60		
IMPINGEMENT DISTANCE FROM FACE, IN.		COMBUSTION STABILITY DEVICE	ACOUSTIC RESONATOR INTEGRAL WITH CHAMBER AND LINER
FUEL, IN.	0.030		
OXIDIZER, IN.	0.055	CHAMBER LENGTH, IN.	9.0
INJECTOR AND CHAMBER DIA, IN.	3.0	TOTAL WEIGHT, LBM	18.8

* DATA FOR 25K LBF THRUST AT 6.0 ENGINE MIXTURE RATIO

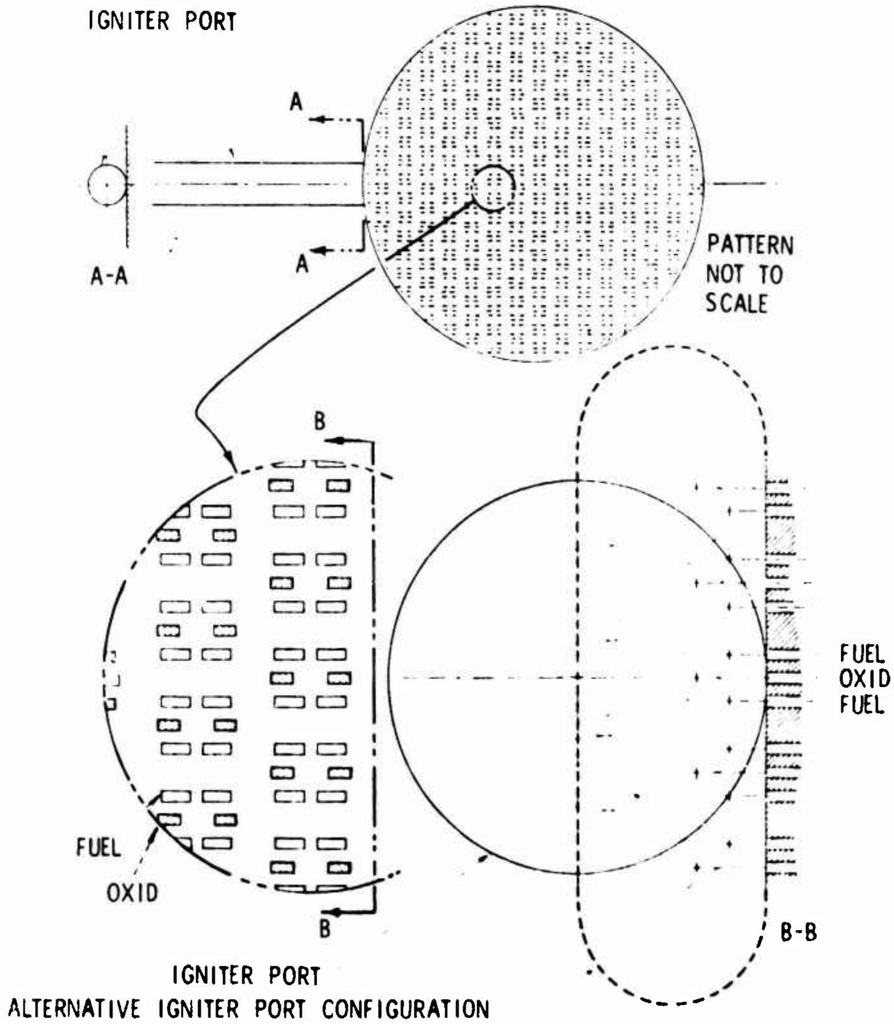


Figure 178. OOS Preburner Injector Pattern

The preburner weight is 18.8 lb, not including the dual redundant electrical exciter/spark electrode assembly, and is listed in Table XXVI with the other major combustion component weights.

The thermal and structural analyses of the preburner are in Appendices B and C.

e. Thrust Chamber Igniter

The thrust chamber igniter design selected for use with the OOS engine is shown in Figure 179. It is a torch type, spark-initiated device which supplies hot gases through a "flame tube" to the center of the injector face. The design requirements for the OOS O_2/H_2 ignition system include that it must be capable of 1500 space restarts. A spark-initiated torch chamber concept is recommended for the OOS thrust chamber ignition system on the basis of ALRC's experience during the last two years with this type of O_2/H_2 igniter on the Space Shuttle Auxiliary Propulsion System (APS) engines, and more recently, during the SSME Phase B testing. Test experience during this time period has shown the spark torch igniter concept to be highly reliable with long life potential.

Many ignition concepts other than the selected, torch type, were considered for the OOS engine. Many of these alternative concepts were investigated prior to the development of spark-torch APS igniters. Before selecting the spark-initiated torch concept for the APS engine, a trade-off study between possible ignition concepts was made. A summary of existing O_2/H_2 ignition energy sources is listed in Table XXXIX. The spark ignition technique was selected; it has been successfully used on the RL-10 and J-2 O_2/H_2 engines. Also, it is capable of producing minimum ignition delay, on the order of 10 milliseconds. The catalyst ignition technique was not selected because of its low response and sensitivity to mixture ratio. The plasma arc technique was not selected because it is not yet considered to be a state-of-the-art device. The hypergolic fluid injection technique was not favored due to its toxicity and handling problems.

Given an ignition energy source, there are two basic concepts by which it can be applied to ignite the rocket engine. The first is the torch or pilot burner concept, in which the ignition source ignites a small portion of the total flow which in turn ignites the remainder of the flow. The second is the direct ignition concept in which the ignition source is applied directly to the main flow. The torch ignition concept was selected for the APS application because it appeared to be more controllable and hence, more reliable than direct ignition. Also, it was desirable to have the capability of using the igniter as a minimum impulse bit thruster. Although the torch concept is considered the prime candidate, a direct ignition concept was also considered for the OOS application and found to offer significant reductions in weight and potential savings in development time. It was, therefore, decided that the direct ignition concept, illustrated in Figure 177, be utilized in the single chamber preburner, and the torch type with the main thrust chamber.

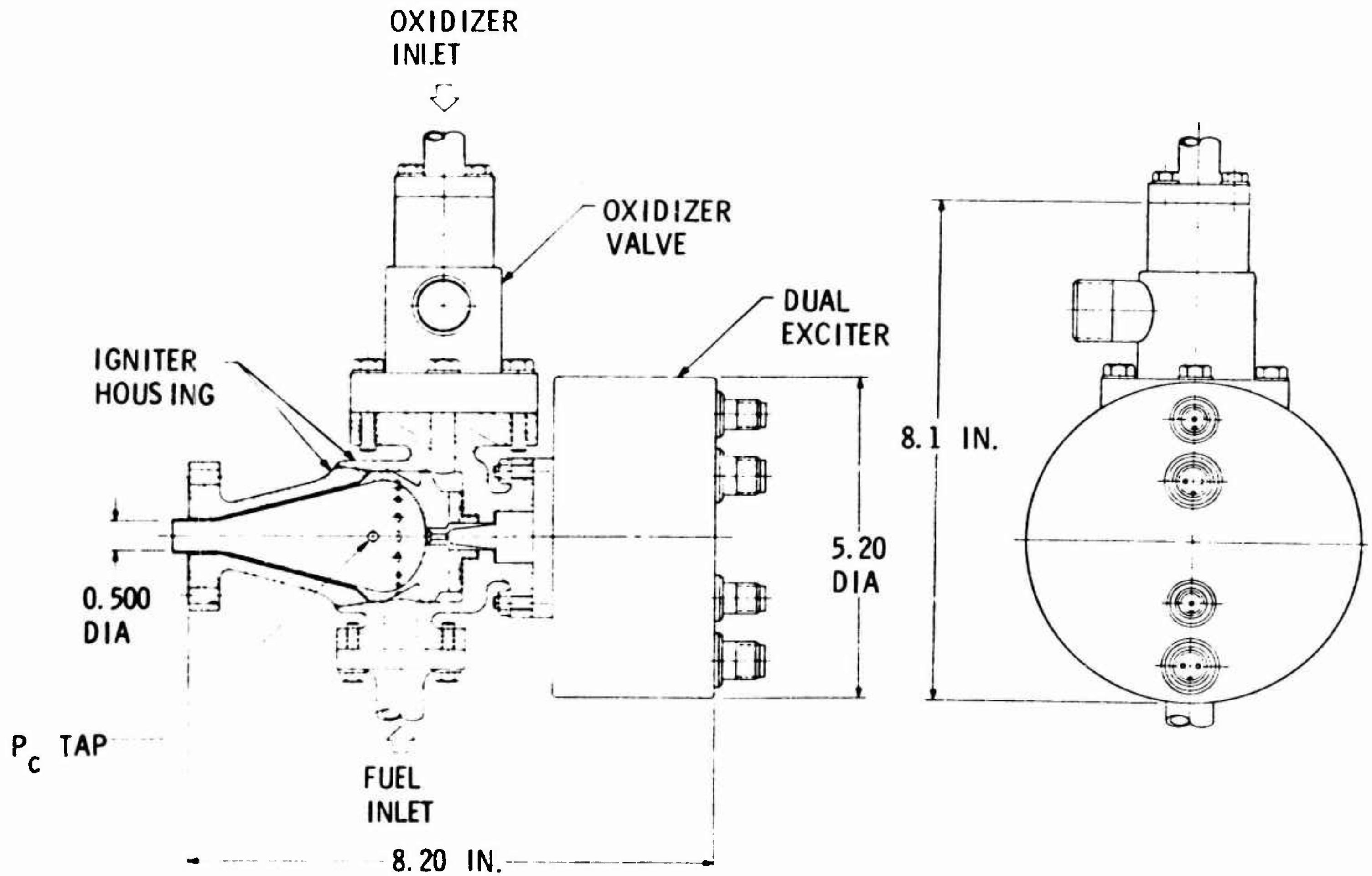


Figure 179. Thrust Chamber Igniter

TABLE XXXIX

H_2/O_2 IGNITION SOURCES

Ignition Device	Development Status	Restraints Placed on Overall Systems	Anticipated Development Problems	Expected Maintainability	Advantages
Spark	State-of-the-art (J2 RL 10)	Oxidizer Lead External Electrical Circuit to Control Spark	Electrode Cooling Flame Quenching at Low Pressures and Temperatures	Good	Potential for High Response
Catalyst	State-of-the-art	Fuel Lead at Ignition and Lag at Shutdown	Cat. Bed Control. Icing of Bed During Space Shutdown. Poor responce at Low Temperatures	Periodic Replacement of Catalyst	No external Power
Plasma Arc	N/A	High Electric Power Requirement	Electrode Durability	Periodic Replacement of Electrode	Insensitive to Propellant Inlet Variability
Hypergolic Injection	State-of-the-art	Separate Feed System and Controls Ground Handling Support Problems	Compatibility	Good	Potential For High Response
Adiabatic Compression	N/A	High Pressure Gas For Driving Piston	Piston Seizing at Low Temperatures May not Ignite at Low Temperatures	Good	Simplicity

In the direct ignition concept, the ignition electrode is placed directly in the injector face of the chamber being ignited. In this application, oxygen is ducted to the electrode from the injector manifold. Hydrogen reaches the electrode by diffusion. This concept used in the OOS preburner effects a weight savings in ignition chamber and valve weights of about 12 lb. Development time may be shorter than that required for the torch igniter since a significant portion of the igniter development is attributed to the torch chamber life requirements. The direct ignition concept was not selected for use in the thrust chamber because of space limitation and expected high maintenance requirements.

OOS Torch Igniter Design Description

The OOS thrust chamber igniter is shown in Figure 179, and its basic design specifications are listed in Table LXXL. The torch igniter is supplied with hydrogen and oxygen from the main propellant system; the propellants are ignited in an ignition chamber by an electrical spark discharge. The resulting hot combustion gases are directed to the main injector, where main chamber ignition occurs by transfer of energy between the igniter exhaust gas and the main injector flows.

The ignition system consists of an ignition chamber, an integral spark plug assembly, an oxidizer valve, and appropriate flow control orifices. Dual spark exciters, contained in the same housing, are used for redundancy. The spark plug is an air-gap type with an internal ceramic-to-metal seal capable of withstanding 10,000 psi internal pressure, as well as severe pressure and temperature cycling. This unit is bolted to the ignition chamber for ease of maintenance. The exciter boxes deliver a peak voltage of 20KV at the electrode tip. The spark rate of each exciter is 50 cps, and the spark energy is 5 millijoules per spark.

The torch igniter consists of an injector head, an uncooled chamber liner, and a pressure shell. These components form a brazed and welded unit to minimize weight and eliminate leakage paths. The injector head contains both oxidizer and fuel injection orifices and manifolds. Fuel enters the injector head through a side inlet and is distributed radially through an annulus at the periphery, and injected tangentially into the ignition chamber through twenty 0.0505 inch dia. orifices. Tangential injection is used to provide film cooling for the injector head and chamber liner. The oxidizer is injected into the ignition chamber through an annular orifice located in the center of the injector head. The annular orifice is formed by inserting the high voltage spark electrode into the center of the oxidizer orifice. The orifice hole is 0.270 inch in diameter and the electrode tip is 0.200 inches in diameter, thus forming a 0.035 inch spark gap. The oxygen protects the spark electrode from the combustion gas environment, and maintains it at sufficiently low temperature that, no electrode tip erosion is encountered with prolonged use. The oxidizer-rich environment at the spark

TABLE XL

THRUST CHAMBER IGNITER BASIC DESIGN SPECIFICATION

Type	Hot Gas Torch
Initiator	Spark
Electrode Cooling	Submerged in O_2 Flow
Spark gap width, in.	0.035
Spark Voltage, kv	20
Spark Rate, Sparks/sec	50
Spark Energy, millijoules/spark	5
Torch Mixture Ratio,	1.5
Torch Temperature (Minimum), °F	1100
Fuel Flow Rate, lbm/sec	0.022
Oxidizer Flow Rate, lbm/sec	0.033
Duration of Operation, sec/Engine Start	0.75
Chamber Pressure (Minimum) psia	7.5
Materials:	
Housing	Armco 22-13-5
Chamber Liner/Tube	Haynes 188
Electrical Seal	Brazed Metal/Ceramic
Injector Head	Zirconium-Copper

discharge is conducive to reliable ignition; this principle of the submerged electrode has proven highly successful on the APS ignition and thruster contracts, NAS 3-14348 and NAS 3-14354, as well as in the SSME Phase B main engine igniter experimental program.

The injector head is made of zirconium copper to provide uniform temperature distribution in the injector head. The chamber liner is a 0.100 in. thick shell made of Haynes 188. It is brazed to the injector head and is positioned 0.010-0.015 in. away from the wall of the pressure shell to provide a stagnant gas insulation to minimize heating of the pressure shell and attachment flange. The chamber pressure shell is fabricated from ARMCO 22-13-5. The igniter is attached to the preburner or hot gas manifold with eight, high strength bolts. The upper section of the pressure shell is also made of ARMCO 22-13-5 stainless steel.

The igniter gases are ducted from the main injector igniter to the center of the injector through an uncooled duct as shown in Figure 179. The duct extends from the igniter exhaust nozzle down through the distribution plate and exits flush with face. The duct is made from Haynes 188.

<u>Torch Igniter Flowrate and Mixture Ratio Requirements</u>

The igniter flowrate and mixture ratio (MR) requirements are influenced primarily by the start transient and ignition requirements. The selected torch MR is 1.5, the total flowrate being 0.055 lb/sec. Although the ignition requirements exert the strongest influence, the transient operating conditions must be a satisfactory durability standpoint. The duration of required torch operation during the initial start transient conditions therefore limits the mixture ratio range within which the igniter may operate. For a given igniter duration, the torch MR during the start is critical because it could cause igniter burnout if the MR is too high. For this reason, it is desirable to maintain the igniter MR at as low a value as is consistent with good ignition characteristics. Another approach to this problem would be to use a cooled liner in the igniter chamber and gas distribution ducts, however, this greatly complicates the igniter design and is a less desirable approach. The resulting MR requirements are met by placing liquid flow control orifices upstream of the igniter, the orifices being sized for operation with tank head supply pressures.

There are two primary ignition considerations: ignition of the torch chamber, and ignition of the main injector with the igniter exhaust gases. The primary factor involved in ignition of the torch chamber at low pressure is that of flame quenching, which is controlled by the size of the ignition chamber, the mixture ratio at the spark electrode, and the cold flow pressure in the ignition chamber. The factors which control ignition of the main injector (and preburner) are: the ignition delay time, which is controlled by the igniter exhaust temperature; mixture ratio; and the mixing between the igniter gases and engine flows, which is controlled by the igniter exhaust momentum. Each of these items are discussed below, and are applicable to both the thrust chamber and preburner assemblies.

The factors which determine the required minimum cold flow pressure for ignition is the igniter chamber size and the mixture ratio at the spark electrode. The OOS igniter chamber diameter is selected as 1.8 inches on the basis of the ALRC SSME research igniter chamber evaluated during the Phase B Program. The preburner chamber diameter is more favorable, being 3.0 inches and utilizes the similar type of igniter flow injection. A series of ignition tests were run with the research igniter to determine the minimum required igniter cold flow pressure as a function of igniter mixture ratio and to demonstrate the igniter concept. A plot of the ignition data obtained with the prototype igniter is shown in Figure 180. Also shown are the flame quenching ignition limits for a 2.0-in. diameter chamber using homogeneous mixtures of oxygen and hydrogen. The data shows that the prototype igniter will ignite and burn at mixture ratios well below that predicted for homogeneous mixtures. The reasons for this are that the igniter uses a unique oxygen submerged spark electrode which ensures an oxygen-rich environment at the spark discharge. Also, the injector is designed with a nonuniform mixture ratio distribution at the injector head such that locally higher mixture ratios than the overall exist, which sustains combustion even at low overall mixture ratios. Testing with the submerged spark electrode igniter has shown that it will ignite immediately (within 20 ms) upon introduction of the oxygen no matter how long the fuel lead (this is because the mixture ratio at the spark discharge immediately becomes oxidizer-rich). Low pressure ignition has been shown to be reliable at fuel-rich mixture ratios down to 0.26. Figure 180 indicates that a minimum cold flow pressure of about 3.5 psia is required for the selected mixture ratio of 1.5. The flowrates were selected to provide a minimum cold flow pressure of 7.5 psia to ensure reliable ignition.

The factors to be considered in selecting the igniter mixture ratio are the minimum required ignition temperature and the chamber material temperature tolerance. The minimum ignition temperature for oxygen/hydrogen mixtures is about 1500°R (1040°F). Temperatures somewhat in excess of this are desired to allow for variations in mixture ratios. A mixture ratio of 1.5 was selected, which produces a gas temperature of 2800°R.

Preburner Flammability Limits and Direct Ignition System

The preburner is designed to have the capability of operating over the wide range of mixture ratios and chamber pressures, as tabulated in Figure 181. While the planned operating mixture ratios are above the flammability limits for H_2/O_2 as indicated in Figure 181, the design allows for operation above that level to ensure high power level capability at extreme conditions. Successful operation at these low mixture ratios was obtained by the ALRC SSME preburner, whose operating conditions are also indicated in Figure 181. The reason that below flammability limit combustion can be sustained is that the burning is initiated in the locally higher mixture ratio that exists within the igniter flame and at the injector face. The locally higher mixture ratio combustion generates sufficient temperature and free radical concentrations to sustain combustion at the lower mixture ratios. Therefore, to sustain combustion at mixture ratios below the

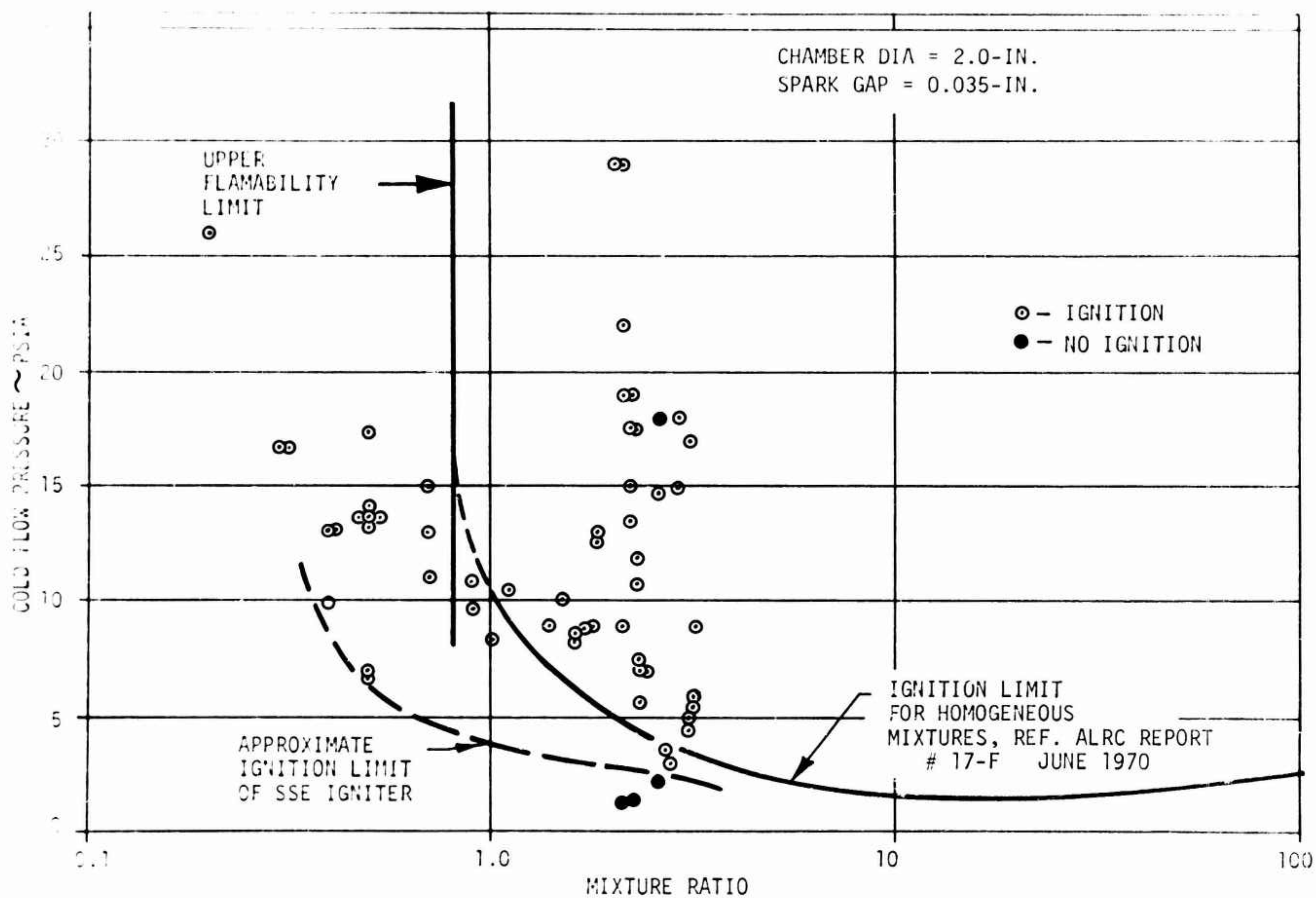


Figure 180. Ignition Limit Data - SSE Prototype Igniter

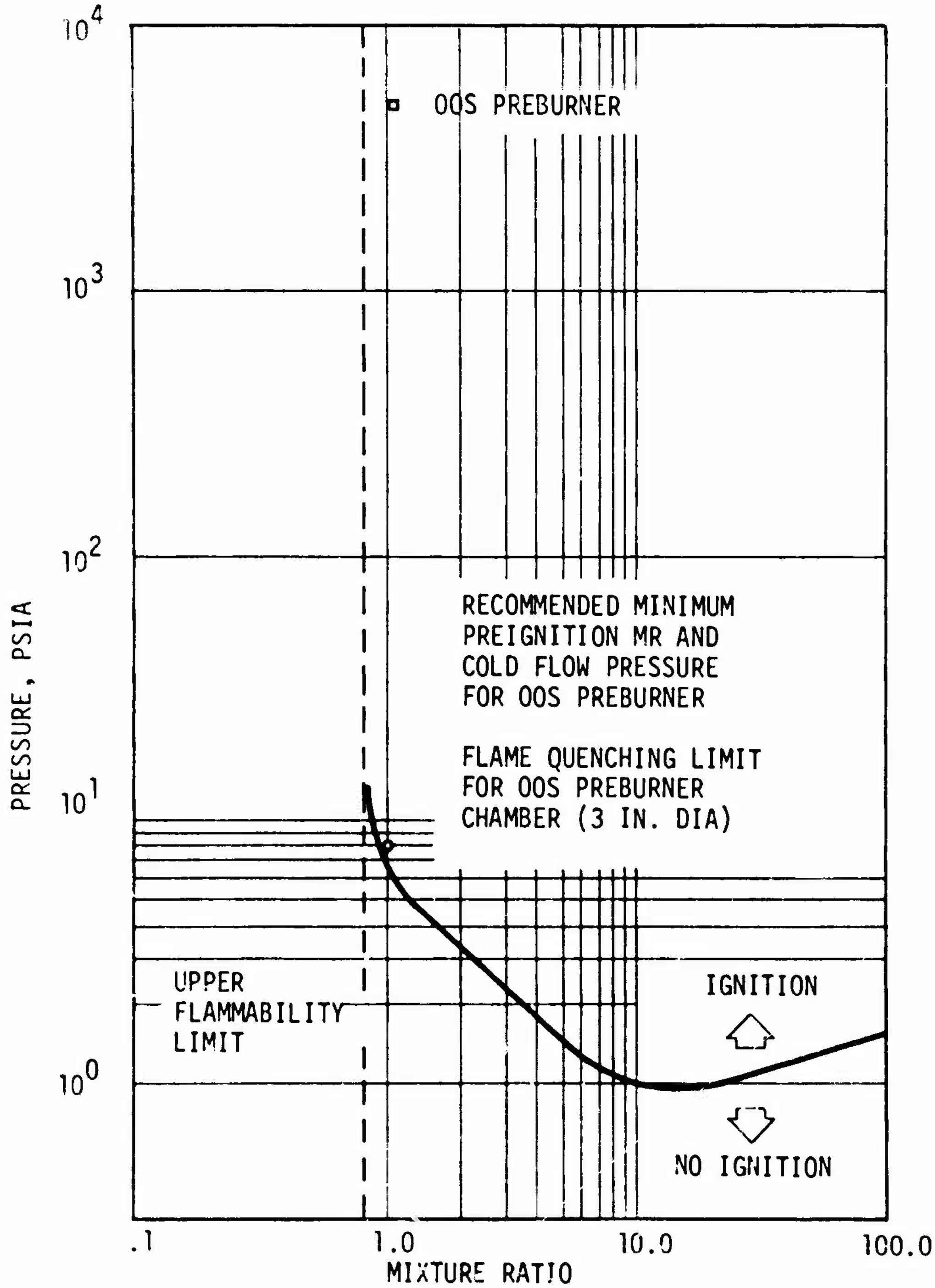


Figure 181. Flammability Limit for Direct Ignition System

flammability limit, it is imperative that combustion occur in the recirculation zone near the injector face prior to the completion of mixing.

The factors which determine whether or not combustion can be sustained are the recirculation gas residence time, temperature, and the local mixture ratio. The criteria for flame-out prevention is that the ignition delay time within the recirculation zone must not exceed the gas residence time. This criteria is illustrated in Figure 182.

The ignition delay time was calculated for each of the OOS preburner operating points. The ignition delay times were calculated assuming that the recirculation gas temperature is equal to the equilibrium combustion gas temperature determined by the overall MR. Calculations were made for assumed local mixture ratios equal to the overall MR and equal to stoichiometric MR. The assumed stoichiometric MR is probably most realistic since combustion is going to occur on the oxidizer jet periphery. The results of these calculations are indicated in Figure 183. It is seen that the OOS preburner will flame-out at minimum power levels when the preburner MR is reduced below 0.4 to 1. The minimum MR planned is 0.6 to 1. The preburner direct ignition system is discussed in Section III,2,d.

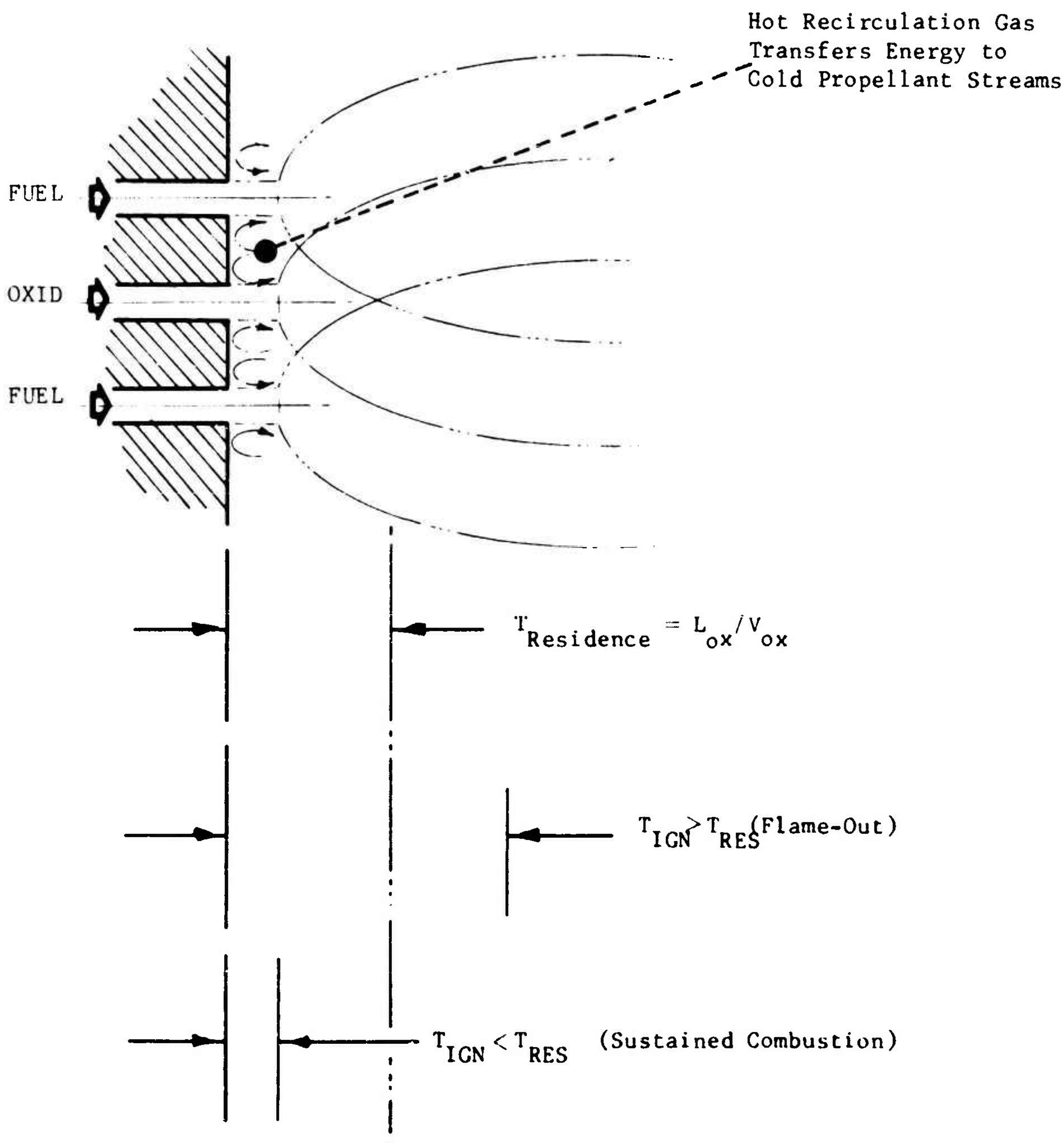


Figure 182. OOS Preburner Flameout Criteria

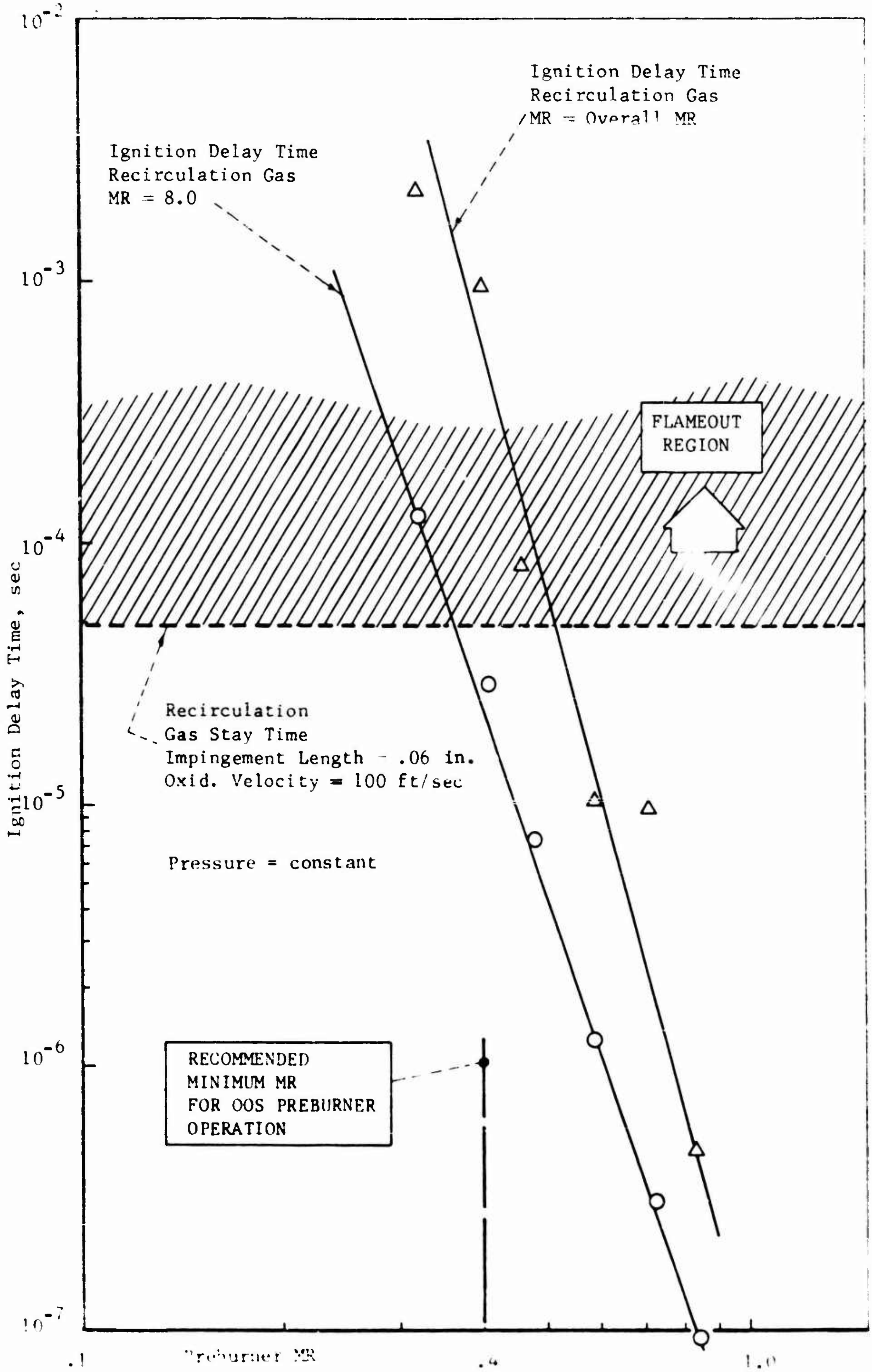


Figure 183. OOS Preburner Flameout Limit

f. Control Valve Concept

(1) General

The valve design requirements were compiled from two sources; Contract F04611-71-C-0040, and those derived from system studies. Included in the contractual criteria are; long term space storability, long life, reusability, low maintenance, man-rating, fail-safe components, multiple restarts, and low cost. In conjunction with the customer's requirements, ALRC determined the following controls criteria; size, leakage rates, concept, response time, system pressure balance, start and shutdown modes, flow rate distribution, and actuation method. A summary of the design criteria for the engine controls is presented in Table XLI.

The OOS Engine is required to throttle over a 5:1 range with a mixture ratio variation of 5.5 to 6.5. The type of thrust and MR control and the system location of these components were determined by computer runs at the nominal full-thrust level, as described in Section III.B.1.c. Considered in this optimization were control valve effectiveness, component pressure differential, interaction between modulating components, and the control location within the system. The primary selection to achieve engine control is a combination of liquid modulating preburner oxidizer and oxidizer pump discharge valves. This method of control avoids the development problems associated with hot gas valves, if used for controlling turbine speed. Other major system valves are a fuel pump discharge shutoff valve and a fuel start bypass valve. The bypass valve regulates flow through the engine chamber jacket during chilldown and idle mode. A modulating turbine bypass valve is located in the fuel circuit to provide desired flows at partial engine thrust.

Controls weight and envelope were considered in establishing the engine pressure balance. By assigning relatively high pressure differentials to each component, the flow diameter can be minimized. This results in minimum envelope dimensions with reduced component weight. The turbopump weight increase, to allow higher component ΔP, is insignificant compared to component size and weight with a low ΔP requirement.

(2) Controls Configuration Selection

The selection of each control configuration resulted from an analysis of the application requirements and the characteristics of each potential valve type. This analysis considered the advantages and disadvantages of each configuration and weighed them against the specific engine requirements to allow an optimum component selection. Of the design factors considered in the selection, some are more important than others. This is due to the type, size, and mission criteria of the particular engine design. For example, on a single start engine, shutoff seal wear may not be a critical design factor. However, on a multiple start engine, seal wear and the resultant seal degradation with each actuation would be weighed heavily in

TABLE XLI

ENGINE CONTROLS DESIGN CRITERIA

COMPONENT	TYPE	FLOW MEDIA	FLOWRATE LBS/SECOND	PRESSURE DIFFERENTIAL, PSI	FLOW DIAMETER, INCH	OPERATING PRESSURE, PSIA	SUPPLY TEMPERATURE, °R	SUPPLY PRESSURE DURING OPENING, PSIA	SUPPLY PRESSURE DURING CLOSING, PSIA	RESPONSE, SEC.	FAIL SAFE CRITERIA	LIFE CYCLE
Fuel Pump Discharge Valve	Shutoff	LH_2	7.68	60	1.2	4255	91	50	750	.500	Fail Closed	1500
Oxidizer Pump Discharge Valve	Modulating (5 to 1 Ratio)	LO_2	46.06	180	1.2	3092	165	40	1500	.250	Fail Closed	1500
Preburner Oxidizer Valve	Modulating (7 to 1 Ratio)	LO_2	6.3	90	0.5	4424	185	100	4161	.250	Fail Closed	1500
Fuel Start Bypass Valve (Turbine Bypass Valve Same Config)	Modulating (5 to 1 Ratio)	GH_2	0.250	7	1.2	4161 23*	38	23	23	.500	Fail Closed	1500

*For sizing at start condition.

configuration selection. Primary consideration for the OOS components selection was given to pressure differential, life cycle, weight, envelope, sealing characteristics, development risk, reliability, commonality, maintainability, experience level, and modulation capability where required.

A modulating function is required on four of the five major control valves during engine operation. This requirement is considered secondary to that of providing positive sealing shut off. The philosophy being that it is more difficult to provide positive sealing at cryogenic temperature than it is to regulate propellant flow. Therefore, an optimum shutoff configuration was selected and then an analysis was conducted to determine the preferred method of adapting it to a modulating capability.

(3) Design Description

(a) Areas of Design Commonality

The areas of commonality within the controls designs are described separately. These areas of design commonality are; valve configuration, main shutoff sealing, valve shaft sealing, and modulation.

1 Valve Configuration

The shutoff sealing configuration selected for all the major components is the poppet valve. Primary importance in making this selection was given to minimum seal wear over a high cycle life. There is minimal rubbing motion by a translating poppet on its shutoff seal which would result in wear and deterioration. Other concepts, ball and butterfly, result in greater wear with resultant low cycle life. Methods of lifting the seal away during opening have been tried (cam lifted concept); however, the weight increase and added complexity are not advantageous. Other advantages to the poppet valve are minimum leakage, short stroke for fast response, ease of fabrication, and simplicity. Consideration was also given to the flexibility of meeting engine envelope requirements. The angle relationship between the poppet valve inlet and outlet flanges may be varied and the actuator positioned parallel or in-line with the valve element. A summary of the rationale leading to the selection of this configuration is shown in Table XLII.

2 Valve Shutoff Sealing

Poppet shutoff sealing is accomplished with a metal-to-metal configuration. The seal is a thin flexible metal shell, part of the poppet assembly, contacting a conical seat. Being flexible, it conforms to the roundness of the seat and provides a microscopic lapping action between the sealing surfaces. Sealing load remains constant because of controlled deflection. Other cryogenic applications that have successfully used this approach are the NERVA Reactor Cooldown Valve, the Transtage Fluorine Valve, and the NASA Space Storable Valve. ALRC test data indicates that

TABLE XLII

VALVE SHUTOFF ELEMENT EVALUATION

CONCEPT	ADVANTAGES	DISADVANTAGES
Poppet	Minimum seal wear due to no rubbing motion; short stroke for fast response; can achieve zero leak shutoff; simplest most commonly used of all valve types; expands into conventional, inverted, and coaxial poppet configurations; minimum seal wear results in high cycle life and reliability; provides flexibility in meeting envelope requirements.	Pressure loss through valve is relatively high; depending on pressure balance actuation force requirement may be high; requires translating shaft sealing.
Ball	Minimum pressure loss because of straight through flow path; two seals may be used for redundancy; low actuation force requirement; has rotary shaft sealing; low inertia and low actuation force assist in minimizing response time.	Rubbing seals have high wear resulting in low cycle life and reliability; because of rubbing seal leak tightness is not reliable; not normally used in sizes less than 1.0 inch; cam lifted seals have been developed but they add to the valve complexity.
Butterfly	More pressure loss than ball because of gate in flow path; rotary shaft seals offer a design advantage; compact housing; low actuation force requirement.	Difficult to obtain positive seal consistently; rubbing motion produces seal wear; gate, shaft, and bearing loads increase with pressure precluding high pressure applications.

leakage rates of less than 0.1 sccs of helium are attainable using a phosphor bronze seal. Use of a polymer seal was considered; however, the potential of high temperature resulting from engine heat soakback made it an unfavorable choice. Heat soakback could cause seal cold flow with a resultant decrease in the required sealing load capability. Other considerations which make the metal seal advantageous are reusability and longer storage life.

3 Valve Piston and Shaft Sealing

Valve dynamic piston and shaft sealing is accomplished by a wedge-spring loaded Delta Seal manufactured by the Rudolph E. Krueger Company. This design employs a Delta Seal as a high pressure shaft or piston seal in conjunction with an in-series, low pressure bellows seal. The cavity between the two seals is connected to an engine low pressure point. Low pressure bellows attaining high cycle life are within the current state-of-the-art.

4 Valve Modulation

The method of propellant flow modulation selected is the contoured poppet. Advantages of this concept include; minimum overall valve length and weight, simplicity, and the ease of shaping the poppet to engine transient requirements. The rationale for the contoured poppet selection is presented in Table XLIII.

(b) Component Design Description

1 Fuel Pump Discharge Valve

A conceptual drawing of the fuel pump discharge valve is shown in Figure 184. This component has a 1.2 in. flow diameter to meet the requirement of flowing 7.68 lb/sec of liquid hydrogen. The designed pressure differential across the valve is 60 psi. The layout presented in Figure 184 is a fail-operational, fail-safe configuration with a redundant electrical system. This redundancy is not a program component requirement, however, it is intended to be used as a comparison with the other control components which are shown as nonredundant. Figure 185 also presents the fuel pump discharge valve but with a nonredundant actuator; and without the low pressure bellows shaft seal. The redundant electrical motor in Figure 184 is positioned parallel to the valve element and drives the valve shaft through an idler gear and ball-screw arrangement. The change in valve length of approximately one in., because of the added low pressure bellows shaft seal, can be noted in the two drawings. The configuration selected for the OOS Engine would be a combination of the two valves utilizing a single in-line motor with a low pressure bellows shaft seal. The total valve weight would be 8.61 lb with an overall length of 11.25 in. A table comparing valve weights and envelope dimensions, between redundant and nonredundant valves, is presented in Table XLIV.

TABLE XLIII

MODULATING ELEMENT EVALUATION

CONCEPT	ADVANTAGES	DISADVANTAGES
Translating Sleeve	Convenient arrangement for incorporating an integral shutoff capability; can be used for high ratio of flow control (> 50 to 1); can be readily used where flow splitting is required; window shape can be easily changed to meet engine transient requirements.	Requires additional valve length when combined with shutoff element; added sleeves add to valve weight; sleeve sections adds complexity to valve shutoff element.
Contoured Poppet	Requires minimum additional valve length; adapts to poppet shutoff element with minimum weight increase; shape of poppet can be changed to meet engine transient requirements; contoured poppet allows inlet/outlet flange relationship to be varied for envelope flexibility.	Poor flow control in nearly closed position with flow control ratios greater than 50 to 1; additional poppet length requires sliding support.

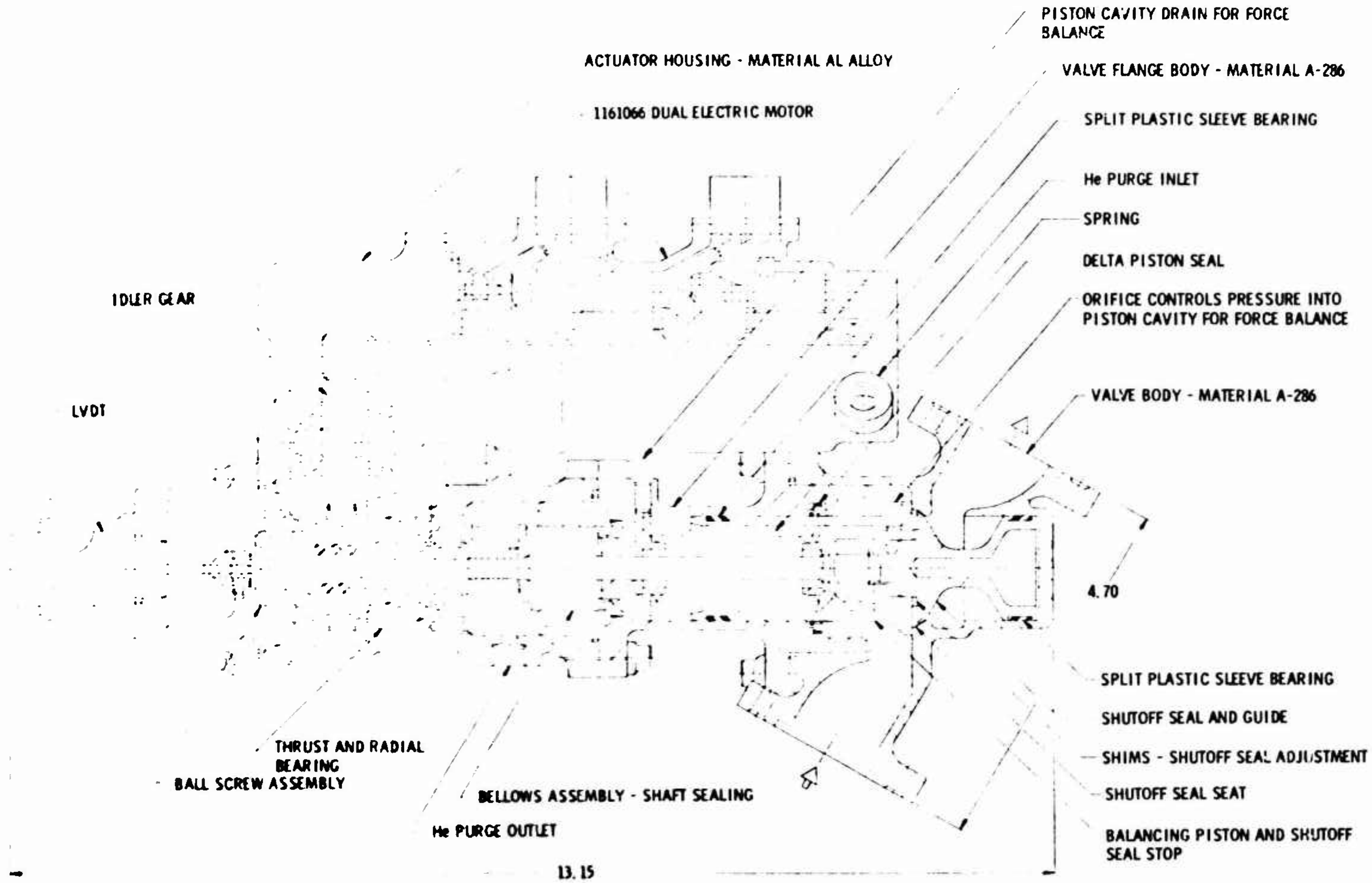


Figure 184. Fuel Pump Discharge Valve, Redundant Actuator

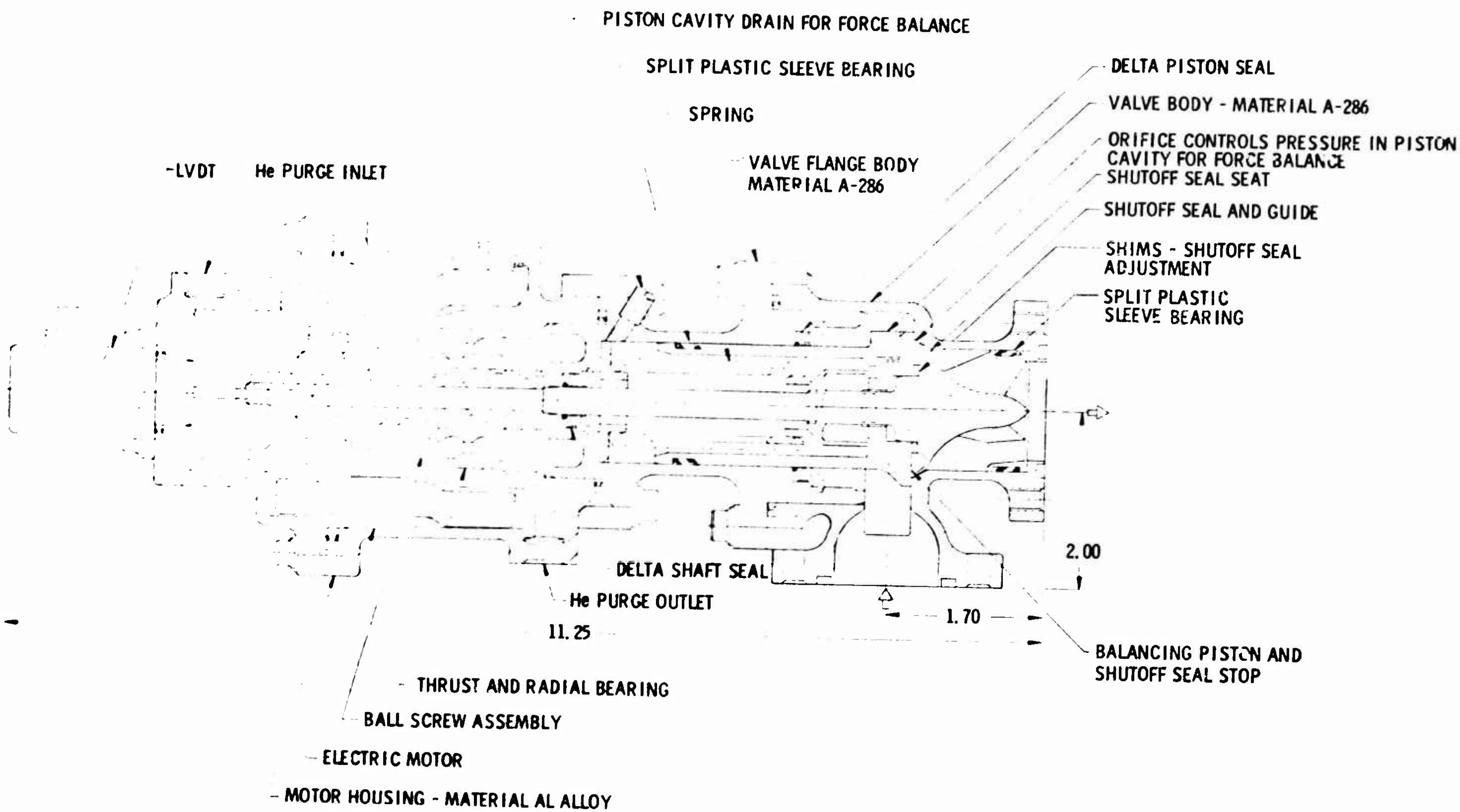


Figure 185. Fuel Pump Discharge Valve, Non-Redundant Actuator

TABLE XLIV

COMPONENT WEIGHT COMPARISON -

REDUNDANT AND NON-REDUNDANT

Component	Valve Weight With Non-Redundant Electrical System, lb	*Valve Weight With Redundant Electrical System, lb	Valve Weight Less Electrical Actuation, lb
Fuel Pump Discharge Valve	8.61	12	5.22
Oxidizer Pump Discharge Valve	9.25	12.64	5.86
Preburner Oxidizer Valve	3.58	6.08	1.08
Fuel Start Bypass Valve (Turbine Bypass Valve Same)	9.25	12.64	5.86

*Redundant weights include gear train weight. Motors would be parallel to valve element which requires an idler gear ball screw arrangement.

The valve will be required to open against an inlet supply pressure of 50 psia and close when the pump discharge pressure is approximately 750 psia. In the event of an actuator failure, the valve return spring will close the valve at a predetermined pressure. In the final design, the force balance conditions will be controlled by the valve pressure balance piston and the closing spring.

2 Oxidizer Pump Discharge and Fuel Start Bypass Valves

The poppet shutoff sealing element of the oxidizer pump discharge valve is the same as the fuel pump discharge valve. Internal dynamic sealing is accomplished by the Delta seal on the valve balancing piston and the low pressure bellows. This valve is located downstream of the oxidizer TPA and performs a shutoff and modulation function between the TPA and the engine injector. The flow diameter is 1.2 in. to meet the requirement of flowing 46 lb/sec (LO_2) with a pressure differential of 180 psi. A drawing of this component is presented in Figure 186. Total valve length is 13.15 in. and the weight is 9.25 lb. A comparison of weight differences between a redundant and nonredundant electrical system is shown in Table XLIV.

Engine mixture ratio is controlled by this valve. Regulation of propellant flow, 46 to 9.2 lb/sec, is accomplished by the contoured poppet which can be varied to meet the required engine start transient.

The oxidizer pump discharge valve is required to open with an inlet pressure of 40 psia and close against a pressure of 1500 psia. The balancing piston orifice, shown in Figure 186, will be sized to allow valve closure at a specified pressure in the event of an actuator failure. Closure is then accomplished by the valve spring force and the unbalanced condition in the closed direction.

The fuel start bypass valve is a modulating valve used during a prestart chilldown and the early part of a start transient (or idle mode). It is used to control the gaseous hydrogen flow through the main chamber cooling jacket. This component is located between the downstream side of the fuel pump discharge valve and the fuel TPA turbine exhaust line. The maximum gas flow required is 0.250 lb/sec with a pressure differential of 7 psi. The flow diameter required is 1.2 in. and is the same as the fuel and oxidizer pump discharge valves. Conceptual drawings of the fuel start bypass valve turbine bypass valves are not included because they will be the same configuration as the oxidizer discharge valve.

3 Preburner Oxidizer Valve

The function of the preburner oxidizer valve is to regulate liquid oxygen flow to the preburner for thrust control. Positive shutoff capability is also required. The maximum flow rate is

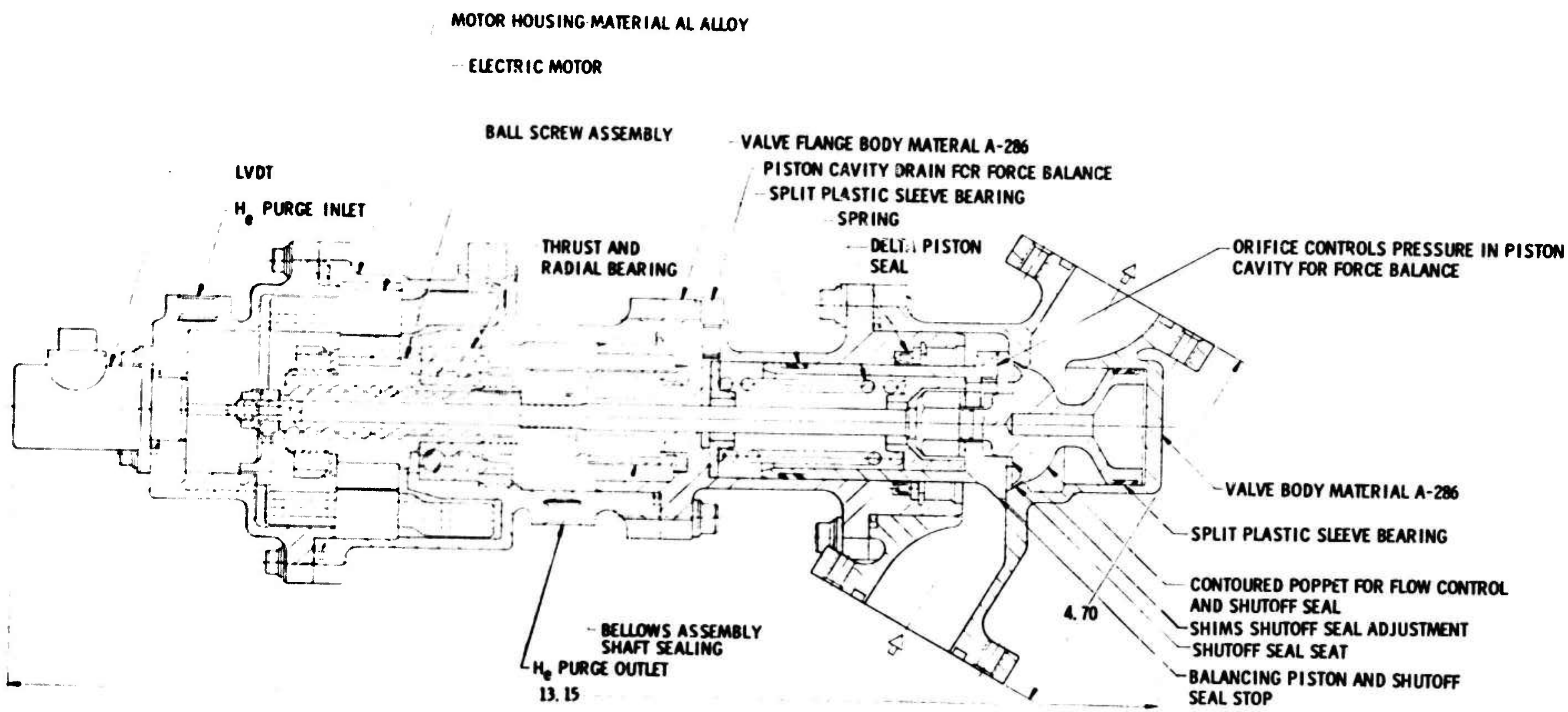


Figure 186. Oxidizer Pump Discharge Valve

6.3 lb/sec with a pressure differential of 90 psi. Minimum flow rate, in a throttled mode, is 0.9 lb/sec. The flow diameter at the maximum conditions is 0.5 in. A conceptual drawing of this component is shown in Figure 187. The configuration of this component is the same as the other valves however it is smaller because of the flow requirements. The valve overall length is 9.00 in. and the weight is 3.58 lb.

4 Igniter and Purge Valves

Two similar igniter assemblies are used on the engine, one for the preburner and the other on the main engine. An oxidizer control valve is an integral part of each igniter assembly. These valves are normally spring-loaded closed and open-signal initiation to the valve solenoid. The coaxial type poppet valve was selected for the following reasons: (1) All leak paths can be eliminated except for the main valve seat and static interface seals. (2) With coaxial valves of 0.375 in. size or less, smaller size and less weight can be achieved than with other solenoid valves due to the placement of the coil around the flow passage. (3) The only moving part required with a coaxial valve is the poppet.

If fail-operational, fail-safe, is required, a secondary solenoid coil would be added. The secondary coil would function in the event of primary coil failure.

System purge requirements have not been defined. Purge valves, nominally 0.25 and 0.50 in., are considered to be within the present state-of-the-art.

(c) Materials Selection

Selection of materials of construction for the OOS valves was primarily governed by the operating environment, propellant compatibility, material properties, structural stability, fabricability, and life cycle reuse. Table XLV lists some of the materials selected. A-286, an iron base superalloy, was selected for the valve shafts and bodies. The selection of A-286 was based on its machinability, adequate strength for the valve operating pressures (max. 4200 psia), and nonsusceptibility to hydrogen embrittlement.

(4) Electromechanical Actuators

The major engine control valves are actuated by a single basic type electric motor. Advantages and disadvantages of hydraulic, pneumatic, and electrical actuation systems are listed in Table XLVI. The

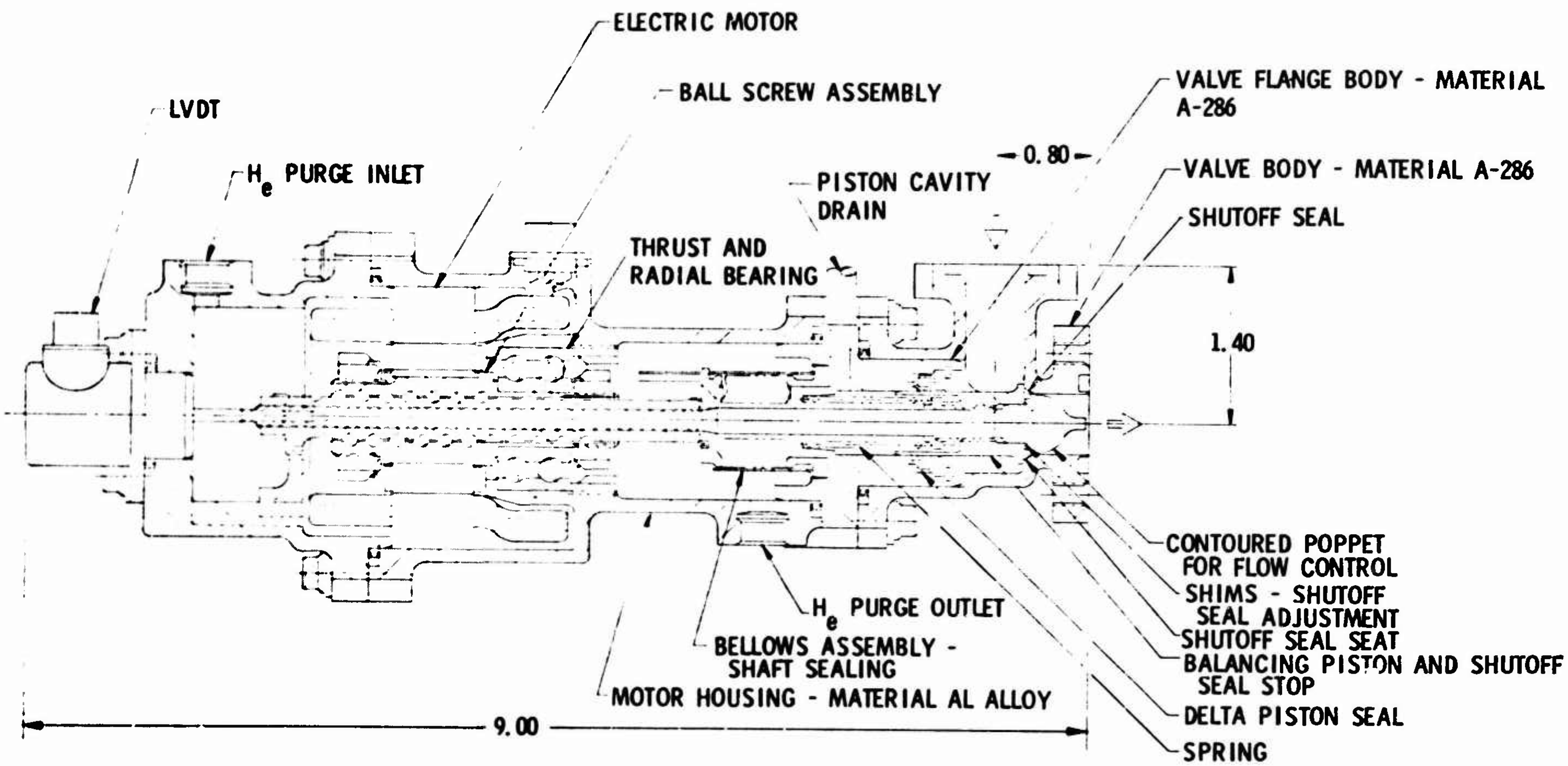


Figure 187. Preburner Oxidizer Valve

TABLE XLV

COMPONENT MATERIALS LIST

*Component	Material
Valve Bodies	A286
Shafts	A286
Bellows	Nested Type, CRES 304L
Shutoff Seals	Phosphor Bronze Seal on CRES 347 (Electrolized) Seat
Dynamic Shaft Seals	15% Graphite-Filled Teflon
Guide Bushings	Teflon
Valve Springs	CRES 302 Spring Tempered
Electric Motor Housings	356-T6 Aluminum Alloy

*Note: Only major subcomponents are listed.

TABLE XLVI

COMPARISON OF HYDRAULIC, PNEUMATIC, AND ELECTRICAL ACTUATION SYSTEMS

System	Advantages	Disadvantages
Hydraulic		
1. Propellant	No third fluid required; actuator integrated into valve; minimum envelope required.	Repeatability affected by bleed-in on first actuation; requires propellant overboard venting; sensitive to system pressure transients; actuator piston seals may be a problem at propellant temperature; requires engine propellant usage.
2. Motor Driven Hydraulic	Well developed system; no wasted fluid; no bleed-in problems; repeatable; unlimited cycles; high pressure actuation; actuator integrated into valve; ease of functional checkout.	High system weight; large number of moving parts; large envelope required; continuous motor operation increases electrical power requirements; long term storage effects not known; requires third fluid; component temperature changes affect repeatability.
3. Gas Over Hydraulic	Permits use of high pressure actuation without motor pump; reduced electrical requirement compared to motor driven hydraulic; other advantages the same as motor driven hydraulic.	Response time varies dependent on number of cycles as a function of reservoir pressure; high pressure gas required; overboard venting of fluids required; additional gas source components required; complex system.
Pneumatic	Light-weight; compact; fast response; components well developed; no bleed-in problems; actuator integrated into valve; minimum electric power required.	Response time varies as a function of number of cycles (gas source pressure decay); repeatability affected by gas temperature; functional checkout requires supply of actuation gas; number of cycles limited by supply volume.
Electrical	Not subjected to contamination (contaminates within a pneumatic or hydraulic system); ease of checkout; can be readily adapted to redundant system; unlimited cycling; long storage life; no control valves required; minimum moving parts result in low maintenance and high reliability; vehicle interface is electrical connection only; readily controls a modulating valve.	Electrical motor systems are not normally used where fast response is required; if transmission is required then complexity increases; total electrical power required exceeds that of hydraulic or pneumatic systems; cycle life depends on gear trans.

coupling to the valve element will depend on envelope requirements. Where overall valve length is not a constraint, the motor may be attached to the end of the valve element. The driver in this case would be a ball-screw arrangement. However where minimum length is required, the motor will be parallel to the valve which will necessitate an idler gear. A parallel arrangement is shown in Figure 184.

The motor selected to drive the actuator is a three-phase, 400 cycle squirrel cage induction type. These motors are rugged, not complex, high performing, and are operated with controlled frequency. They are superior to other types of brushless motors when low cost and reliability are prime considerations. A comparison of electric motor configurations is listed in Table XLVII. Brushless dc and ac synchronous motors were studied. The brushless dc motor requires permanent magnets on the rotor and a motor mounted communtating device which creates additional failure modes.

The motor selected to drive all but the preburner oxidizer valve would have a nominal design torque of 20 in.-lb. The smaller preburner oxidizer valve would be driven by a motor with a nominal torque of 10 in.-lb. A detailed analysis during a final design phase may indicate that it would be advantageous, from a cost-development standpoint, to use the larger motor on all control valves. A table listing the valve and actuator force requirements is presented in Table XLVIII.

(5) New Technology Areas

There are no areas within the controls concepts, considered for the OOS Engine, that may be classified as new technology. There are, however, certain areas of new technology which in the future could effect the OOS engine component designs. These areas would include: (1) A material that has compliance and resilience at cryogenic temperatures for use as shutoff seals, dynamic shaft seals, and static seals. (2) A readily castable material having good low temperature characteristics for high pressure applications. An ideal material for this application is Inconel, however, it cannot be reliably made as a casting. (3) Lighter weight materials having high magnetic properties for use in electrical actuator motors. This would result in smaller and lighter motors for a given power output.

TABLE XLVII

COMPARISON OF ELECTRIC MOTOR CONFIGURATIONS

TYPE	ADVANTAGES	DISADVANTAGES
Brushless DC Torque Motor	Fast response; ease of checkout; good repeatability; proportional control capability; low power drain.	Large size; heavy weight; permanent magnet in motor requires special handling to prevent demagnetization; generally used in low torque applications; will not allow spring closure of valve in the event of power loss caused by shorted stator windings.
3-Phase Induction Motor	Allows valve closure (spring) with loss of motor power for any reason; operates with controlled frequency; simplest of all motor configurations; high experience level; minimum size and weight; high reliability and low cost.	Complex signal controller required.

TABLE XLVIII

VALVE AND ACTUATOR FORCE REQUIREMENTS

Component	Pressure Start of Opening, psia	Pressure Start of Closing, psia	Response Time, sec	Valve Force, Max., lb	Actuator* Linear Force Output, lb	Max. Current Amps (Closed to Open)	Max Load HP
Fuel Pump Discharge Valve	50	750	.500	287	600	11	1.43
Oxidizer Pump Discharge Valve	40	1500	.250	465	600	11	1.43
Preburner Oxidizer Valve	100	4161	.250	240	400	7	0.76
Fuel Start Bypass Valve (Turbine Bypass Valve Same Config)	23	23	.500	150	600	11	1.43

*Two size motors, 10 inch-lb and 20 inch-lb, are utilized in this study. The 10 inch-lb torque motor is used on the preburner oxidizer valve.

g. Turbopump and Low Speed Boost Pump

(1) Requirements

The turbopumps for the 25,000 lbf vacuum engine must be capable of functioning within specification for the service life without overhaul of at least 10 hours with a capability of 300 starts.

In addition to the service life requirement, the turbopumps must be lightweight, capable of delivering LO_2 and LH_2 at NPSH values of 16 and 60 feet, respectively, and also be capable of delivering propellant to the engine over a throttle range of 5:1 mixture ratio range of 5.5:1 to 6.5:1.

(2) Design Selection

(a) Main Fuel Turbopump

High impeller tip speeds and high shaft rotational speeds are necessary to produce high fuel pressure (4255 psia) with a minimum number of stages at an acceptable efficiency. High tip speeds cause high stresses in the impeller discs, and since ductility decreases at cryogenic temperatures, fracture toughness characteristics establish impeller tip speed limits. High rotational speeds constrain bearing sizes due to DN values, and shaft dynamics and critical speed considerations limit the design speed. Additionally, high turbine efficiencies require high rotor tip speeds which result in high disc stresses. The turbine must also operate at relatively low inlet temperatures in order to obtain the required material strength. Most high nickel, high temperature turbine materials in a condition of high strain are subject to hydrogen embrittlement when exposed to a high pressure hydrogen environment; consequently, unique design approaches are necessary in the turbine area to achieve the required life and reliability. The high pressures on the impeller and rotor faces, which generate axial shaft loads several orders-of-magnitude higher than the bearing capacity, impose the need for an axial thrust balancer. The design and location of this balancer is largely influenced by the configuration of the pump.

In addition to selecting the best component for the application, the components must integrate into a flightweight package in a manner such that speed and critical speed requirements are met, axial thrust balancing can be achieved, component thermal environments can be controlled, and pressure and temperature deflections and distortions are minimized or can be accommodated. The conceptual turbopump design shown in Figure 188 was established after considering these requirements.

The required pressure rise can be achieved with either a two- or three-stage pump utilizing either open or shrouded impellers. Shrouded impellers were selected over open impellers for the following reasons:

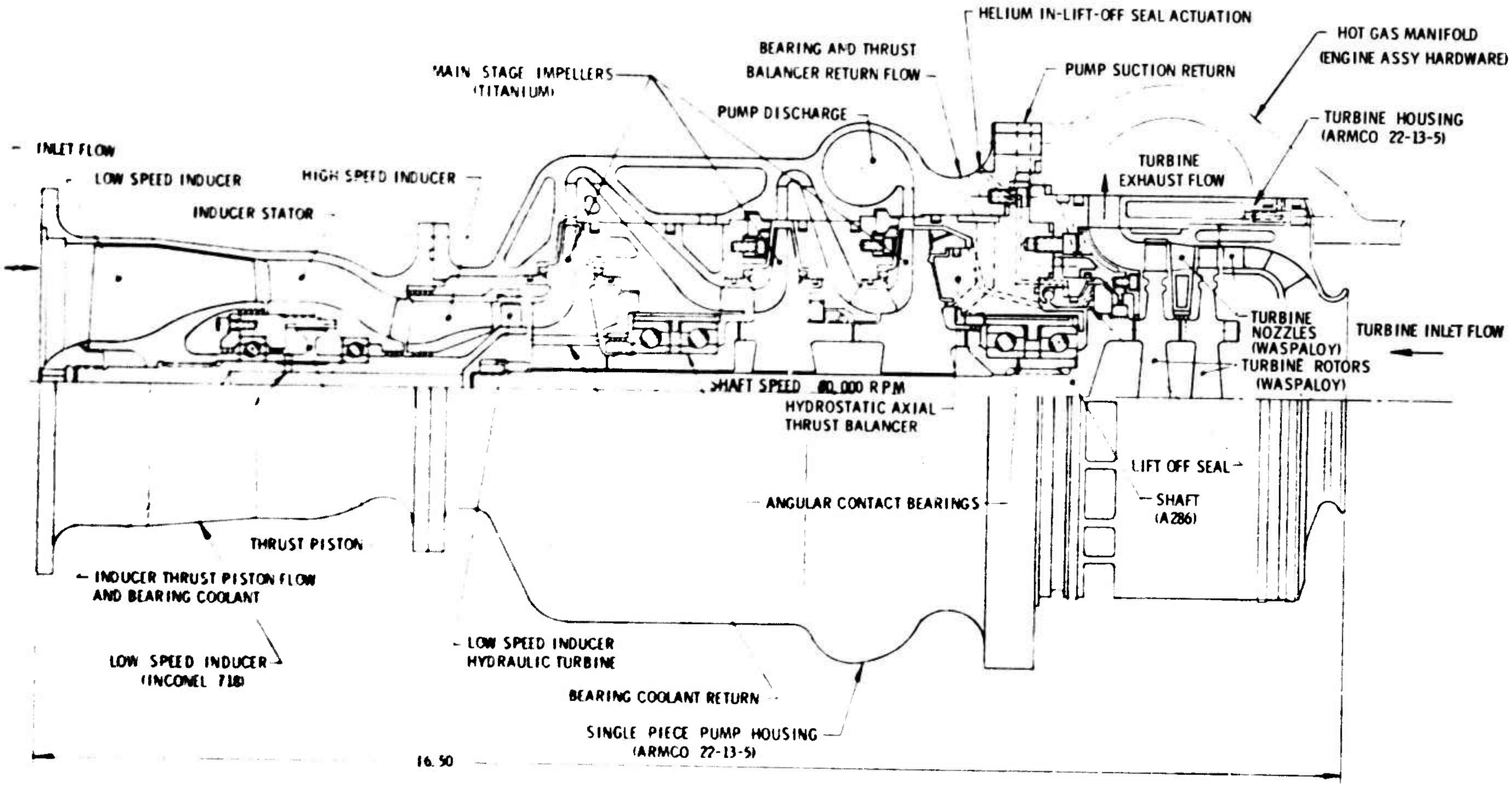


Figure 188. Fuel Turbopump Concept

- More predictable axial thrust balance, less capacity required from thrust balancer, and hydrostatic wear ring diameters easily adjusted;

- Less sensitive to axial clearances: allow reasonable housing distortion and shaft translations without performance or thrust variations; and

- Provide flexibility for component arrangements and means for providing bias load for thrust balancer.

The baseline design for the turbopump is a three-stage pump configuration.

A two-stage pressure compounded turbine was selected since it provides the required horsepower at an acceptable efficiency. A single stage turbine could be used; however, the efficiency would be 12 points lower which would require a higher pressure ratio, thus a high pump discharge pressure. A three-stage design was not considered practical due to critical speed requirements and the design complexity of three stages.

Rolling contact ball bearings were selected for the turbopump for the following reasons:

- The turbopump must have a low breakaway torque;

- Hydrostatic bearings are not considered state-of-the-art in the area of rubbing starts where high radial loads may be present;

- Roller bearings have high capacity and high spring rates, but are susceptible to end wear problems at the DN values required;

- High spring rates are not required for the bearings since the shaft passes through the rigid body criticals and the bearings must be spring mounted; and

- Duplex angular contact ball bearings can be used in an arrangement which provides high radial load capacity and prevents axial thrust balancer rubbing during the start transient.

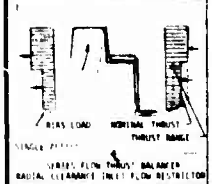
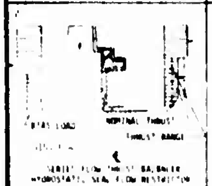
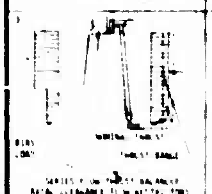
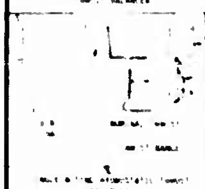
The thrust balancer concept is a single acting design capable of compensating for thrust in a single direction. Several types of thrust balancers were considered before this design was selected. Figure 189 shows five different balancer design considerations and gives advantages and disadvantages of each. The thrust balancer concept selected (configuration 2) has all of the required characteristics and integrates well with the shrouded impellers and into the overall turbopump.

Figure 190 shows four different types of dynamic seals that were considered but not selected for use between the turbine and pump, and the advantages and disadvantages of each are given. The concept selected is illustrated with the TPA design, Figure 188.

1 LH_2 TPA Design Parameters

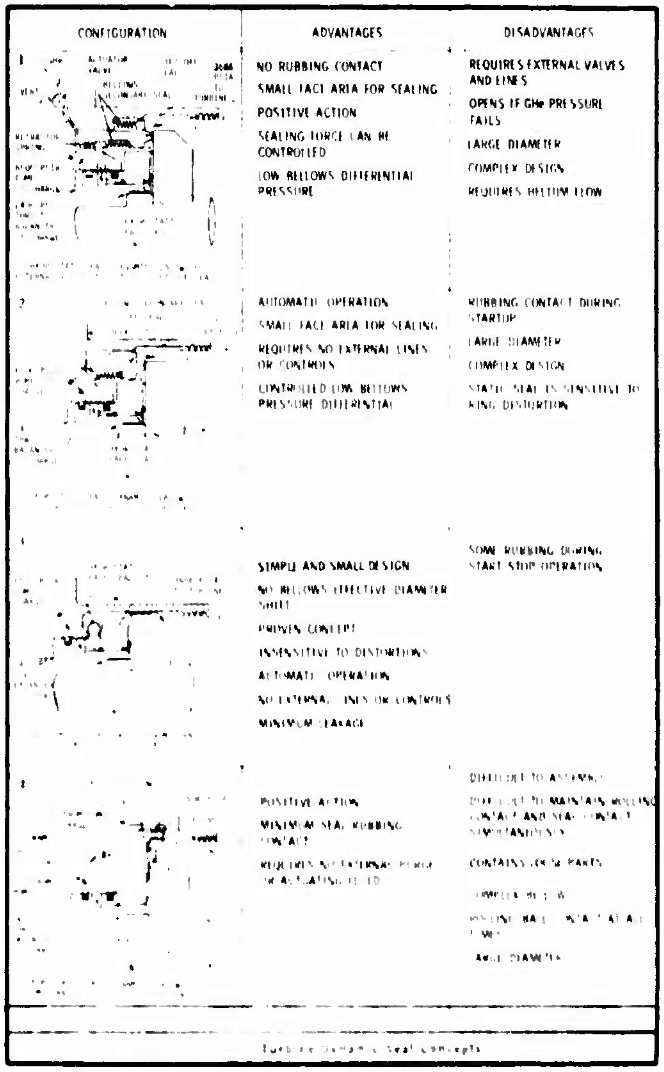
The design parameter for the LH_2 and LO_2 propellant feed turbopumps are given in Table XLIX where design parameters are identified to the component level for the boost turbopumps and the main turbopump. The main turbopumps include the high speed inducer, main pump, turbine and the power transmission section. A discussion of pertinent design parameters is given in the following sections:

The main fuel turbopump design was evaluated with a two-stage and a three-stage main pump. The basic differences in design parameters between the two turbopumps are impeller stresses, pump efficiencies, and design complexity. The two-stage pump has an efficiency value that is approximately four points lower than the three-stage pump; however, the engine cycle can accept this lower efficiency and the reduced mechanical complexity could be considered a desirable tradeoff. The governing aspect, however, is impeller stresses. The two-stage pump must operate with an impeller tip speed of 2017 ft/sec whereas the three-stage pump reduces the tip speed to 1505 ft/sec. A parametric analyses conducted to determine tip speed limits for turbine rotors and pump impellers (Ref. Appendix Report No. SA-OOS-TM-01) for the necessary life cycles of the OOS components shows that pump impeller tip speed is limited to approximately 1600 ft/sec by fracture toughness characteristics which implies that a three-stage pump is necessary. A Titanium 5Al-2.5 Sn alloy impeller has a 1200 cycle to failure predicted life with an initial flaw size of 0.02 in. which is a typical nondestructive test inspection limit. A minimum safety factor of four has been used in this analysis; therefore, a design cycle life of 300 is obtained by dividing the 1200 cycle to failure life by the safety factor 4. The two-stage pump configuration results in an impeller tip speed of 2017 ft/sec. This increased tip speed results from the fact that the pressure rise is developed by two pump stages instead of three and the required pressure rise is 235 psi greater due to the lower overall pump efficiency of the two-stage pump configuration. This tip speed for the same titanium alloy results in a predicted cycle to failure life of 600 (life reduction from over speed not included). This cycle life divided by a safety factor of four gives a 150 cycle design life which does not satisfy the

CONFIGURATION	ADVANTAGES	DISADVANTAGES
1 	LOW FLOW RATE REQUIRED FOR THRUST BALANCER MINIMUM NO. OF PARTS	SENSITIVE TO HOUSING RADIAL DISTORTION PERFORMANCE DEPENDS ON UNIFORMITY OF INLET FLOW RESTRICTOR
2 	MINIMUM INFLUENCE FROM IMPELLER DISTORTION OR ANGULAR MISALIGNMENT LOW FLOW RATE GOOD STABILITY REDUCED SENSITIVITY TO HOUSING DISTORTION	REQUIRES MORE PARTS
3 	SIMPLE DESIGN	SENSITIVE TO IMPELLER DISTORTION POOR STABILITY DIFFICULT TO INTEGRATE WITH BALL BEARINGS LARGE FLOW RATE
4 	POSITIVE CONTROL BOTH DIRECTIONS GOOD STABILITY CAN UTILIZE ALTERNATIVE FLUID SUPPLY	REQUIRES SEPARATE DISK ADDS WEIGHT AND LENGTH TO ROTOR AND HOUSING HIGH FLOW RATE REQUIRED SENSITIVE TO HSG DEFLECTION DIFFICULT TO INTEGRATE WITH BALL BEARING REQUIRES SUPPLY LINES
5 	MINIMUM NO. OF PARTS CAN UTILIZE ALTERNATIVE FLUID SUPPLY STABILITY WHIRLING CRITICAL SPEED UNAFFECTED	REQUIRES SUPPLY LINE HIGHER FLOW RATE THAN SERIES DESIGN ANGULAR RESTORING FORCE AFFECTS ROTOR DYNAMICS THROUGH CRITICAL SPEEDS

Evaluation of Various Thrust Balancer Types

Figure 189. Evaluation of Various Thrust Balancer Types



CONFIGURATION	ADVANTAGES	DISADVANTAGES
1	NO RUBBING CONTACT SMALL FACE AREA FOR SEALING POSITIVE ACTION SEALING FORCE CAN BE CONTROLLED LOW BELLOWS DIFFERENTIAL PRESSURE	REQUIRES EXTERNAL VALVES AND LINES OPENS IF GHe PRESSURE FAILS LARGE DIAMETER COMPLEX DESIGN REQUIRES HELIUM FLOW
2	AUTOMATIC OPERATION SMALL FACE AREA FOR SEALING REQUIRES NO EXTERNAL LINES OR CONTROLS CONTROLLED LOW BELLOWS PRESSURE DIFFERENTIAL	RUBBING CONTACT DURING STARTUP LARGE DIAMETER COMPLEX DESIGN STATIC SEAL IS SENSITIVE TO RING DISTORTION
3	SIMPLE AND SMALL DESIGN NO BELLOWS EFFECTIVE DIAMETER SHIFT PROVEN CONCEPT INSENSITIVE TO DISTORTIONS AUTOMATIC OPERATION NO EXTERNAL LINES OR CONTROLS MINIMUM LEAKAGE	SOME RUBBING DURING START STOP OPERATION
4	POSITIVE ACTION MINIMUM SEAL RUBBING CONTACT REQUIRES NO EXTERNAL PURGE OR ACTUATING FLUID	DIFFICULT TO ASSEMBLE DIFFICULT TO MAINTAIN ROLLING CONTACT AND SEAL CONTACT SIMULTANEOUSLY CONTAINS LOOSE PARTS COMPLEX DESIGN ROLLING BALL CONTACT AT ALL TIMES LARGE DIAMETER

Turbine Dynamic Seal Concepts

Figure 190. Turbine Dynamic Seal Concepts

TABLE XLIX

25K ENGINE TPA DESIGN PARAMETERS

	Oxidizer	Fuel
Boost Pump Assembly		
NPSH, ft	16	60
Thermodynamic Suppression Head, ft	Calc.	Calc.
Min Ratio of Effect. NPSH, Run/Breakdown	1.87	1.87
Suction Specific Speed, Breakdown	45,000	45,000
Max. rpm Ratio, Main/Boost Pump	3.5:1	3.5:1
Specific Speed, Max	4,000	4,000
Suction Dia Ratio, Hub/Tip	0.4	0.3
Tip Dia Ratio, Exit/Suction	1.0	.916
Mean Dia Ratio, Exit/Suction	1.1	1.054
Turbine Drive	FFH*	FFH*
Ratio Bearing Spacing/Shaft Dia	2.17	2.17
Ratio Turbine Overhang/Shaft Dia	3.17	3.17
Bearing Dn., mm x rpm	215,000	343,000
rpm	26,700	
Main High Speed Turbopump Assembly		
High Speed Inducer		
Thermodynamic Suppression Head, ft	Calc.	Calc.
Min Ratio of Effect. NPSH, Run/Breakdown	1.87	1.87
Suction Specific Speed Breakdown	30,000	30,000
RPM Ratio, Inducer/Main	1:1	1:1
Specific Speed, Max	4,000	4,000
Suction Dia Ratio, Hub/Tip	0.512	0.515
Tip Dia Ratio, Exit/Suction	1.0	1.0
Mean Dia Ratio, Exit/Suction	1.1	1.1

* FFH - Full Flow Hydraulic

TABLE XLIX (cont.)

	Oxidizer	Fuel
High Speed Main Pump		
Thermodynamic Suppression Head, ft	Calc.	Calc.
Min Ratio of Effect. NPSH, Run/Breakdown	2.27	2.50
Suction Specific Speed Breakdown	10,000	10,000
Shaft Speed	50,000	80,000
Number Stages	1-1/2	3
Internal Recirculation, allocated, %	5	7
High Speed Main Turbine		
Type	Axial	Axial
Number Stages	1	2
Energy Extraction Means	Impulse	PI**
Inlet Temperature, °R	1,860	1,860
Mean Blade Speed, ft/sec	1,100	1,300
Nozzle Angle	15	20
Max Dia Ratio, Hub/Tip	.90	.90
Min Dia Ratio, Hub/Tip	.85	.85
Inlet Manifold Mach No.	.3	.3
Exit Manifold Mach No.	.5	.5
Turbine Bypass (Allocated % Leakage)	4	4
Power Transmission		
Turbine End Bearing Type	DPLX Ball	DPLX Ball
Turbine End Bearing Dn	1.5×10^6	2×10^6
Pump End Bearing Type	DPLX Ball	DPLX Ball
Pump End Bearing Dn	1.0×10^6	2×10^6

** PI-Pressure Compounded, Impulse

specified 300 cycle life value. Designing the impeller with a lower safety factor value implies designing to failure and also leaves no margin in the design for growth potential. Therefore, the two-stage pump configuration was considered unsatisfactory for the specified thrust chamber pressure schedule and life requirements and the three-stage pump configuration was selected. The fuel turbopump shaft speed was determined to be 80,000 rpm, this being the maximum allowable shaft speed compatible with turbine blade root stress consideration and from bearings life consideration. These limitations are discussed in the turbine and power transmission sections below.

a NPSH Considerations

The selection of design parameters for pumps and inducers when pumping cryogenic propellants such as LH_2 and LO_2 must take into account the marked improvement in cavitation performance exhibited by these propellants over that obtained with room temperature water. This improvement depends upon the physical properties of the pumped fluid, the flow conditions, and the heat transfer effects between the liquid and vapor cavity formed on the leading edge of the inducer vanes. These combined effects are called the thermodynamic effects and generally referred to as the "Thermodynamic Suppression Head" (TSH). The term, TSH, is a convenient way of incorporating the thermodynamic effects into the suction specific speed equation when comparing the performance to that for water in the following manner:

$$S = \frac{NQ^{1/2}}{NPSH^{3/4}} \text{ (WATER) } \approx \frac{NQ^{1/2}}{(NPSH + TSH)^{3/4}} \text{ (PROPELLANT)}$$

Based upon experimental data for LH_2 [(1)], the Thermodynamic Suppression Head (TSH) increases from 90 ft at 34°R to approximately 180 ft at 42°R. For the minimum hydrogen temperature of 38°R, a value 129.22 ft was selected for design purposes.

b NPSH Requirements

A 6.64 psi minimum NPSP is required at the 80,000 rpm main TPA high speed inducer, based on a TPA with no boost pump and the high speed inducer having a suction specific speed capability of 45,000. The breakdown minimum effective NPSH is given by:

$$NPSH_{(EF, B)} = (NQ^{1/2}/S)^{4/3}$$

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(1) NASA Technical Memorandum (TM X-1854) "Cavitation Performance of Line-Mounted 80.6° Helical Inducer in Hydrogen," August 1969

The continuous run minimum effective NPSH is given by:

$$NPSH_{(EF,\ C)} = 1.87 \times NPSH_{EF,\ B}$$

where the 1.87 value represents the minimum ratio of continuous-to-breakdown effective NPSH values to limit cavitation head loss to 2%. The continuous run minimum NPSP can be calculated from the effective NPSH by:

$$NPSH = NPSH_{(EF,\ C)} - TSH$$

$$NPSP = \frac{NPSH \times \rho_{LH2}}{144}$$

A turbopump designed to operate without a low speed pump with a 60 ft NPSH (NPSP = 1.81 psi) would operate at 52,000 rpm and weigh in excess of 200 lb. A turbopump with a 5.3 lb boost turbopump permits the main pump to operate at 80,000 rpm and weigh 32 lb giving a combined weight of 37.3 lb. Further weight reduction of the main TPA by increasing its speed are not feasible since the shaft speed is limited to 80,000 rpm to achieve the specified 10 hr turbine and bearing life.

A low speed pump is utilized to supply the differential pressure between the specified minimum and the minimum required by the high speed main pump. The low speed pump concept selection and its design parameters are discussed below.

The turbopump power balance includes allocated internal flows for bearing cooling and LH_2 leakage from the turbine and bearings into the turbine exhaust. The allocated flows, the predicted values for the design, and the flow control means are presented in Table L.

<u>2</u> Main Fuel Pump

The main pump includes the high speed inducer and three centrifugal stages to develop the required pressure. In addition to developing the required pressure at the rated operating point, the pump is required to operate at flow coefficient values as low as 50% of design when the engine is throttled to 20% of rated thrust. Inducer suction performance is degraded for this reduced flow coefficient operation as shown in Figure 191 for the high speed inducer and the suction of the first centrifugal stage. The design objective for the pump suction is to adjust the design such that the cavitation head loss does not exceed 2%. Pressure oscillations occur in the pump due to swirl instabilities in the inducer when the inducer head loss exceeds the 2% value. Operation with head loss not exceeding 2% can be achieved by operating with an effective NPSH value (NPSH effective = NPSH + Thermodynamic Suppression Head) 1.87 times the breakdown effective NPSH value. It can be seen that the head loss over this throttle range falls below the 2% maximum. The maximum suction specific

TABLE L

LH_2 TURBOPUMP INTERNAL RECIRCULATION FLOW

Flow Location	Predicted lb/sec	Predicted Percent Pump Flow	Seal Type	Percent of Nom. Flow per Stage	Design Allocated Percent
First Stage Imp. Front Side	.180	2.35	Hydrostatic Face Seal	5.07	7% per Stage
First Stage Imp. Back Side	.201	2.62	Hydrostatic Face Seal		
Second Stage Imp. Shaft Seal	.078	0.10	Labyrinth		
Second Stage Imp. Front Side	.193	2.52	Hydrostatic Face Seal	9.06	7% per Stage
Second Stage Imp. Back Side	.184	2.40	Hydrostatic Face Seal		
Third Stage Imp. Shaft Seal	.072	0.09	Hydrostatic Shaft Seal		
Third Stage Imp. Front Side	.193	2.52	Hydrostatic Face Seal	6.57	7% per Stage
Thrust Balancer & Turbine End Brg.	.31*	4.05	Hydrostatic Brg & Labyrinth		
Turbine Seal To Turbine Exhaust	.188	3.4**	Labyrinth	----	4% ** Bypassing Turbine

*Flow can return to suction of second or third stage depending magnitude of hydrostatic thrust force required. Value in total flow for second stage based on return to suction of second stage to achieve max load capability from thrust balancer.

**Percent of LH_2 in turbine flow.

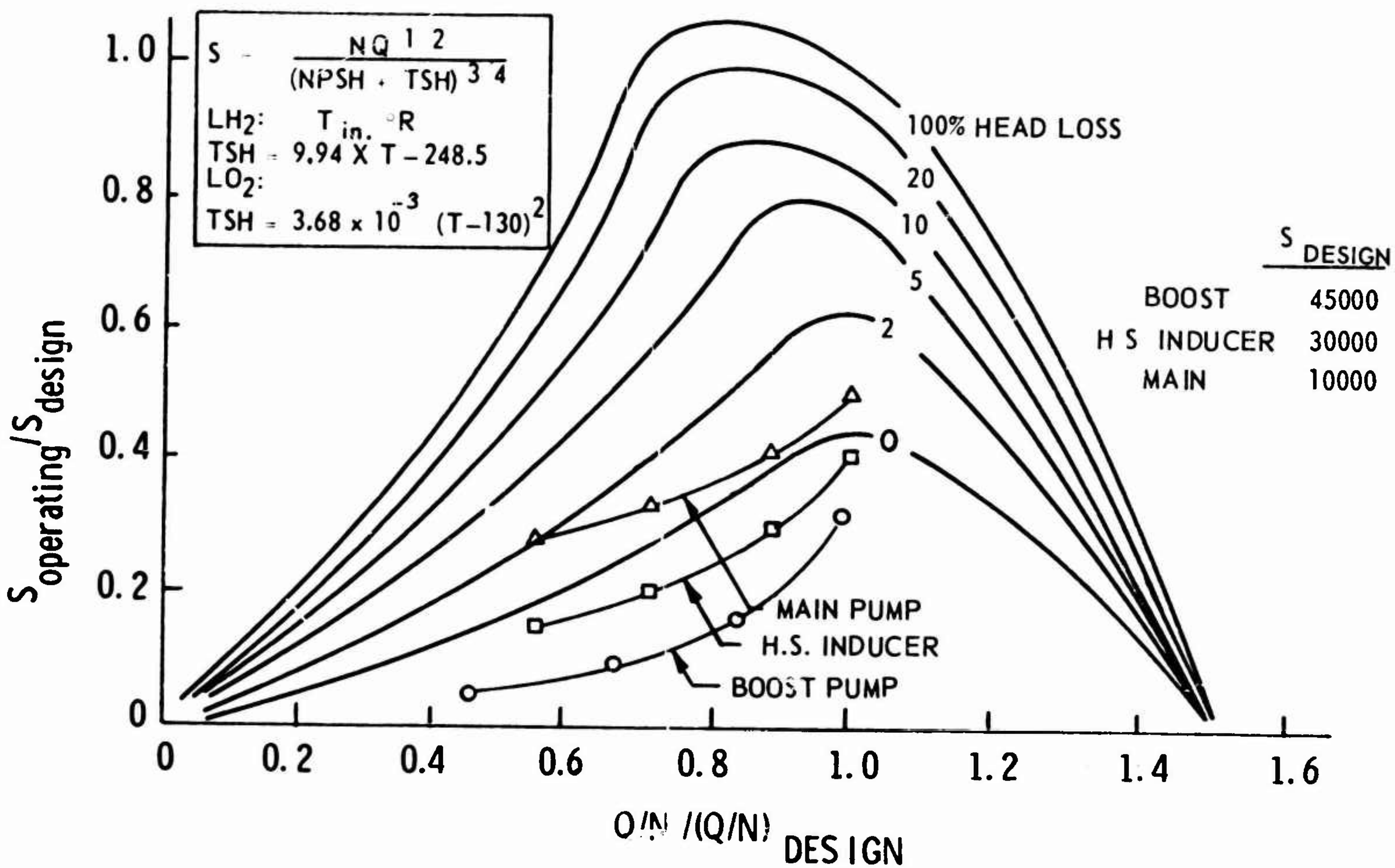


Figure 191. Normalized Fuel Pump Headloss vs Suction Specific Speed and Q/N

speed capability of the high speed inducer at a breakdown was set at 30,000, since the flow to this inducer is leaving diffuser vanes of the low speed boost pump. The high speed inducer maximum suction specific speed operating value is 0.39 times 30,000 = 11,700 as a result of the boost pressure from the boost pump. This delivered pressure from the low speed boost pump is greater than is required to operate at 2% head loss. This results from the design parameters associated with the inducer of the low speed boost pump (see the low speed inducer section below). This excess available NPSH to the high speed inducer results in a ratio of RUN-to-BREAKDOWN NPSH value of 3.36 and the design point S/S_{design} ratio of 0.39.

The inducer suction eye diameter is determined from pump flow, shaft speed, hub-to-tip diameter ratio and suction flow coefficient by:

$$D_{eye} = \left(\frac{\dfrac{93.4 \; \dfrac{Q}{N}}{\phi}}{1 - R_H^{\;2}} \right)^{1/3}$$

where:

Q = GPM

N = RPM

R_H = Hub-to-Tip Diameter Ratio

ϕ = C_m/U, tip flow coefficient

The flow coefficient ϕ is determined from the design breakdown suction specific speed, the design Hub/Tip Diameter Ratio, and an impeller flow incidence to blade angle ratio of 0.425 as shown in Figure 192.

The design maximum suction specific speed capability of the high speed centrifugal pump at breakdown was set at 10,000 (no inducer suction). The NPSH to the high speed main impeller is defined by the discharge conditions of the high speed inducer less the head removed by the full-flow hydraulic turbine used to drive the low speed boost pump. As in the case of the high speed inducer, operation of the centrifugal pump below the 2% head loss line requires an operating effective minimum NPSH 1.87 times greater than the breakdown effective NPSH value. The high speed inducer delivers a pressure greater than is required to satisfy this minimum effective NPSH which results in a ratio of RUN-to-BREAKDOWN NPSH value of 3.1 and a S/S_{design} of 0.50.

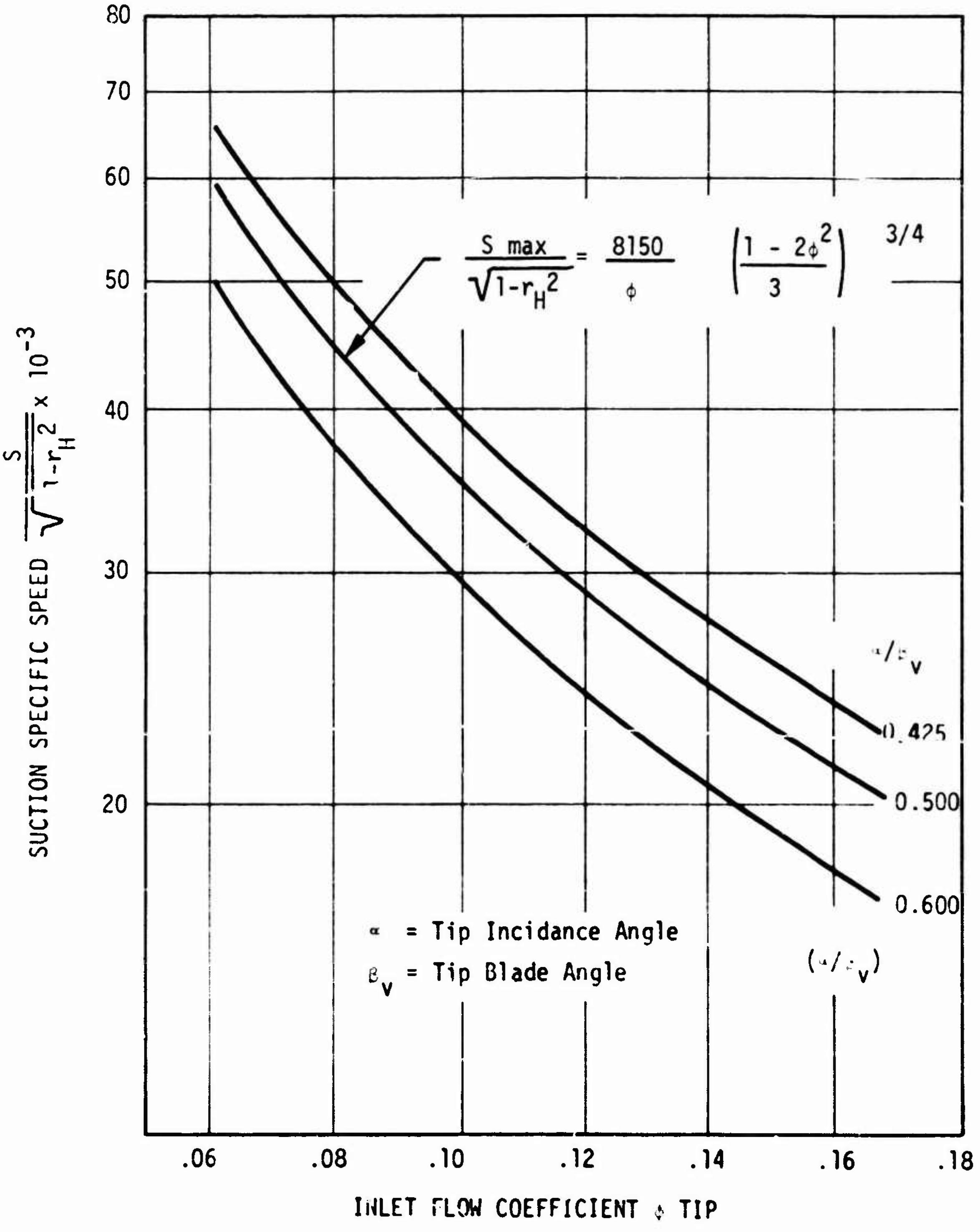


Figure 192. Suction Specific Speed

The pressure rise of the high speed inducer has been shown to be greater than required to drive the hydraulic turbine and satisfy the NPSH requirements of the first centrifugal stage. This excess head generating capability results from limiting the inducers design specific speed to a maximum value of 4000. Designing to a higher specific speed value becomes unrealistic in that the fluid incidence angle to the inducer by itself will result in most of the head generation and a specific speed of 4000. Reduction of the delivered head and increase of specific speed would necessitate reducing the design incidence angle which would de-optimize the suction design point incidence angle necessary to achieve the desired 30,000 suction specific speed.

3 Main Turbine

The main turbine is required to drive the LH_2 pump and utilize the LH_2 which is heated by combustion with LO_2 as a drive means. It is further desired that the pressure ratio across this turbine be the lowest possible value. The required turbine pressure ratio can be reduced by increasing the energy of the gas and increasing the efficiency of the turbine that extracts shaft power. Maximum turbine efficiency values can be achieved with a turbine having a stage blade/gas velocity ratio approaching 0.5. A two-stage (pressure compounded) turbine can approach this value for the magnitude of gas availability required for the stage combustion engine cycle with a chamber pressure of 1800 psia.

A two-stage pressure compounded turbine with impulse blading was selected and is allocated a minimum efficiency value of 65.7%. One single-stage turbine would have eleven points less efficiency than the two-stage turbine and was considered unsatisfactory. A three percent efficiency increase could be achieved with a three-stage turbine. This efficiency increase was considered too low to justify the mechanical complications of a three-stage turbine.

4 Power Transmission

The power transmission portion of the fuel turbopump is required to support the rotating elements of the turbopump assembly and control the internal flow of LH_2. The allocated and calculated values of internal flow and the means for their control are discussed above under LH_2 TPA Design Parameters.

The shaft breakaway torque shall not exceed 12 in.-lb which is 50% of the turbine stall torque developed from a tank pressure start with combustion in the primary combustor. To assure compliance with this low breakaway torque requirement, preloaded duplex ball bearings were selected to support the shaft at the pump and turbine ends. A static lift-off seal was selected to isolate LH_2 from the turbine cavity. Actuation of this seal open prior to engine start eliminates the rubbing friction drag also eliminates the cycle life limitation resulting from wear on starts and shutdown.

The bearings have been designed to have a cryogenic B_1 life of 10 hours which is 0.2 times the life of an oil lubricated bearing. This was achieved by limiting the maximum axial thrust load of the life-limited bearing to 75 lb and the pump radial load to 100 lb (Figure 193). This 75 lb is the summation of 25 lb preload and a 50 lb applied load.

The B_1 bearing life has been reduced by a factor of 0.2 to account for life reduction resulting from cryogenic operation. Experience has shown that wear and not fatigue will establish the usable life of a rolling element bearing operating in liquid hydrogen. Therefore, in the comparative evaluation of various bearing designs, fatigue life alone cannot be used as an absolute criterion for selection. Furthermore, since no fluid film separation of the balls and raceways can be expected, relatively high tractive or tangential surface shear stresses exist and are directly proportional to the normal stresses. These additional surface stresses must be taken into consideration when making life predictions. Results of research programs, conducted by both Tallian[(2)] and Skurka[(3)], wherein rolling contact bearing life was investigated as a function of the dimensionless ratio (lubricant film thickness to raceway surface finish) seem to agree that for operation in low viscosity lubricants where metal-to-metal contact can be expected, lubrication-life multiplier (reduction factor) of at least 0.2 must be taken. Consequently for the purpose of comparative design evaluation, a minimum 0.2 B_1 life of 10 hr was set as a design requirement. A further life reduction would normally be required for the use of AISI 440C stainless steel which is a few points softer than AISI 52100 steel, the base material in fatigue life criteria. However, since a hardness increase will be experienced at cryogenic temperatures, no life reduction for material difference was taken. In tests conducted at Aerojet, the hardness of 440C samples increased from a room temperature hardness of Rc 57 to Rc 63 at -320°F.

(b) Main Oxidizer Turbopump

The main oxidizer turbopump has operating requirements which are less demanding than the fuel turbopump; consequently, the turbopump configuration is more conventional than the fuel turbopump. The limiting design parameters are critical speed, inducer vane stress, and inducer cavitation damage.

The pump design is a "stage-and-a-half" design dictated by engine system requirements. The turbine as shown is a single-stage partial admission design which provides the horsepower at an acceptable efficiency with a minimum of axial thrust. The bearing, thrust

(2) Tallian, T.F.: On Competing Failure Modes in Rolling Contact: ASLE Transactions 10, 418-439 (1967)

(3) Skurka, J.C.: Elastohydrodynamic Lubrication of Roller Bearings; ASME Paper 69-Lub-18

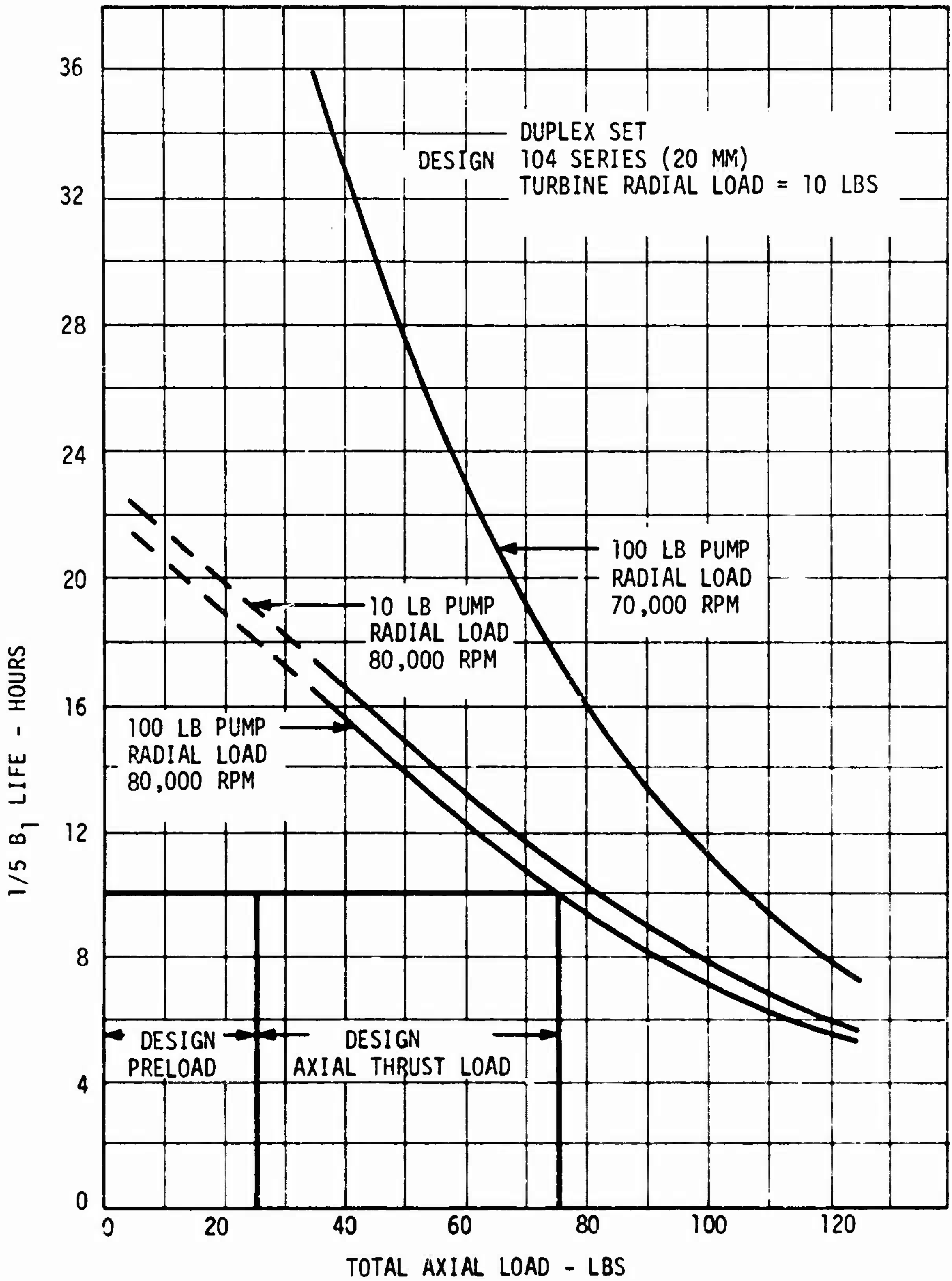


Figure 193. Fuel TPA Turbine End Duplex Bearing Set Life vs Axial Load

balancer, and turbine dynamic seal concepts were all selected for the same reasons as discussed for the fuel turbopump. Several design concepts of the interpropellant seal were considered before the design shown in Figure 194 was selected. The design and operation of this seal system was previously described.

1 LO_2 TPA Design Parameters

The design parameters for the LH_2 and LO_2 propellant feed turbopumps are given in Table XLIX where design parameters are identified to the component level for the boost turbopumps and the main turbopumps. The main turbopumps include the high speed inducer, high speed main pump and turbine, and the power transmission section. A discussion of pertinent design parameters is given in the following section.

The main oxidizer turbopump consists of a high speed (50,000 rpm) pump which is direct driven by a single-stage hot gas turbine. The pump has a main, plus a half-stage, where the main stage pumps all of the LO_2 to the pressure level required at the thrust chamber oxidizer injector inlet. The half-stage takes approximately 15% of this flow and pumps it an additional 1330 psi to the pressure level of the preburner injector. An alternate turbopump configuration was evaluated wherein a single pump stage pumped all of the oxidizer instead of 15% of the oxidizer to the pressure level required by the preburner. The pressure level for the flow going to the thrust chamber was 1900 psi greater than required, this excess pressure being dissipated by a control valve. This single-stage pump concept resulted in an increase of fuel pump discharge pressure to 4850 psi to achieve a turbopump power balance, which is 600 psi greater than required with the main plus half-stage concept.

The one plus half-stage pump concept was selected on the basis of lower fuel pump power balance pressure and system control considerations.

Inconel 718 was selected for the pump impeller. This material operating with a tip speed of the order of 600 ft/sec has a life capability in excess of the maximum specified values of 20 hr and 600 cycles. The oxidizer turbopump shaft speed was established at 50,000 rpm. This shaft speed and turbine sizing adjusted the inertia of the oxidizer TPA such that its effective start time is the same order of magnitude as the fuel TPA start time. The start time values shown in Table LI approximate the time in seconds that the TPA would achieve design speed when the gas generator pressure is applied as a step value to the design pressure. Therefore, the absolute values of start time are not representative of the OOS application since the gas generator bootstraps rather than step-starts, but their relative value show the relative start characteristics from polar moment of inertia considerations.

The NPSP required to satisfy the 50,000 rpm operation is 44.5 psi which exceeds the allocated 7.9 psi (16 feet)

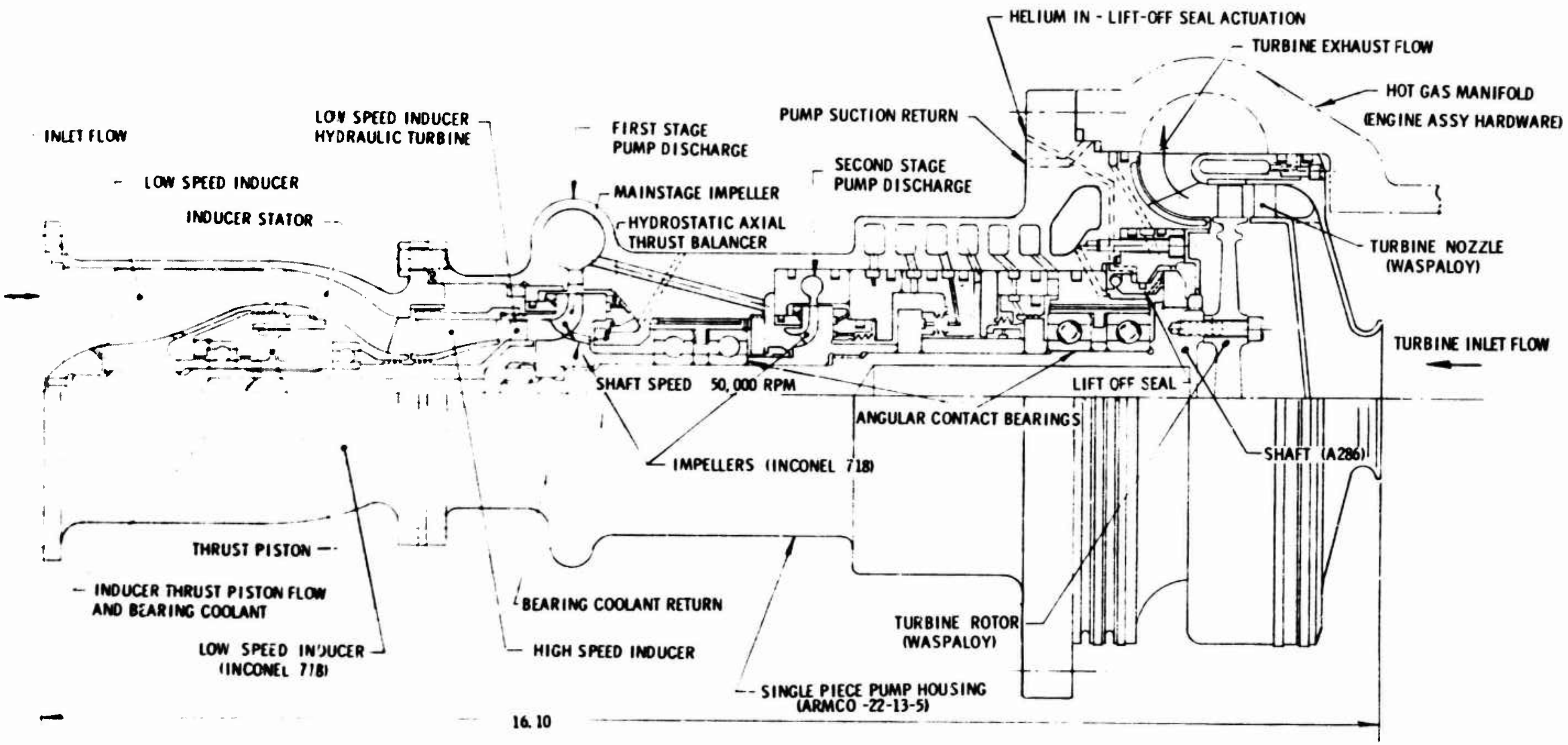


Figure 194. Oxidizer Turbopump Concept

TABLE LI

LO_2/LH_2 POWER BALANCE CALCULATIONS
LO2-LH2

PARALLEL TURBINE

STAGED COMBUSTION CYCLE PROPELLANT FEED SYSTEM

ENGINE PARAMATERS

THRUST, TCA	FTCA	LBF	25000.0000
THRUST, SYSTEM	FSYS	LBF	25000.0000
MIXTURE RATIO, TCA	RMTCA	---	6.0000
MIXTURE RATIO, SYSTEM	RMSYS	---	6.0000
AREA RATIO, TCA	ETCA	---	270.0000
SPECIFIC IMPULSE, TCA	ISTCA	SEC	465.6700
SPECIFIC IMPULSE, SYSTEM	ISSYS	SEC	465.6700
TEMP RISE TCA FUEL COLLANT	DTRGF	DEG R	295.0000
PRESSURE DROP TCA FUEL COOL	DPRGF	PSID	750.5470
CHAMBER PRESSURE, TCA	PCTCA	PSIA	1800.0000

CHAMBER PRESSURE, GAS GENERATOR	PSIA	PCGGO	2972.5318	PCGGF	2972.5318

SYSTEM FLOWS

		SYSTEM		OXIDIZER LEG		FUEL LEG	
THRUST CHAMBER	LB/SEC	WTCA	53.6861	WOTCA	46.0166	WFTCA	7.6694
GAS GEN FLOW, OXID TPA	LB/SEC	WGGO	4.0160	WOGGO	1.8715	WFGGO	2.1445
GAS GEN FLOW, FUEL TPA	LB/SEC	WGGF	10.3465	WOGGF	4.8215	WFGGF	5.5249
TURBINE FLOW OXID TPA	LB/SEC	WGTO	3.8553	WOGTO	1.7966	WFGTO	2.0587
TURBINE FLOW FUEL TPA	LB/SEC	WGTF	9.9326	WOGTF	4.6287	WFGTF	5.3039
SYSTEM	LB/SEC	WS	53.6861	WOS	46.0166	WFS	7.6694

SYSTEM FLOW RATIOS

		SYSTEM		OXIDIZER LEG		FUEL LEG	
MIXTURE RATIO, GAS GENERATORS	---			RMGGO	.8727	RMGGF	.8727
TPA GAS GEN TO TCA	---	RGT	.2675	RGTO	.0748	RGTF	.1927
TPA GAS GEN TO TPA	---	RGP	.2675	RGPO	.0873	RGPF	1.3491
TPA TURBINE TO TCA	---	RTT	.2568	RTTO	.0718	RTTF	.1850
TPA TURBINE TO TPA	---	RTP	.2568	RTPO	.0838	RTPF	1.2951

BOOST PUMPS

PUMPS - AXIAL FLOW		OXIDIZER TPA		FUEL TPA	
WEIGHT FLOW	LB/SEC	WBO	46.0166	WBF	7.6694
VOLUME FLOW	GPM	QBO	290.3173	QBF	791.7912
PROPELLANT VAPOR PRESSURE	PSIA	PVBO	15.0000	PVBF	15.0000
PRESSURE SUCTION	PSIA	PSBO	22.9050	PSBF	16.8100
PRESSURE DISCHARGE	PSIA	PDBO	141.1993	PDBF	43.2175

TABLE LI (cont.)

TEMPERATURE SUCTION	DEG R	TSBO	163.0000	TSBF	38.0000
TEMPERATURE DISCHARGE	DEG R	TDBO	163.4007	TDBF	38.1583
CONTINEOUS RUN MINIMUM SUCTION CONDITION					
NET POS.,SUCTION PRESSURE, MIN CONT. OPER.	PSID	DPSRBO	7.9050	DPSRBF	1.8100
NPSH (CONT. MIN.) ABOVE VAPOR PRESSURE	FEET	HSVPBO	15.9947	HSVPBF	59.9296
THERMODYNAMIC SUPPRESION HEAD	FEET	TSHBO	4.0000	TSHBF	129.2200
NET POS. SUC. HEAD, CONT. MIN. EFFECTIVE	FEET	HSVNBO	19.9947	HSVNBF	189.1496
SUCTION SPECIFIC SPEED, CONT OPER WITH THS	---	SSPNBO	25742.4839	SSPNBF	12610.1488
SUCTION SPECIFIC SPEED CONT. MIN. NO THS	---	SPRXBO	30433.6870	SPRXBF	29860.2266
SHAFT SPEED, PUMP	RPM	RPMBO	14285.6532	RPMBF	22856.9893
BREAKDOWN MINIMUM SUCTION CONDITION					
RATIO RUN TO BREAKDOWN MIN EFFECTIVE NPSH	---	SFACBO	2.1058	SFACBF	5.4533
NET POSITIVE SUCTION PRESSURE, MINIMUM	PSID	DPSBO	2.7158	DPSBF	.0000
NET POS. SUC. HEAD, MIN. ABOVE VAPOR PRES.	FEET	HSVMBO	5.4951	HSVMBF	-94.5344
THERMODYNAMIC SUPPRESION HEAD	FEET	TSHBO	4.0000	TSHBF	129.2200
NET POSITIVE SUCTION HEAD, MIN. EFFECTIVE	FEET	HSVBO	9.4951	HSVBF	34.6856
SUCTION SPECIFIC SPEED, WATER EQUIVALENT	---	SSPBO	45000.0000	SSPBF	45000.0000
SUCTION SPECIFIC SPEED, BREAKDOWN, NO THS	---	SSPXBO	67819.7100	SSPXBF	.0000
PRESSURE RISE	PSID	DPBO	118.2943	DPBF	26.4075
HEAD RISE	FEET	DHBO	239.3524	DHBF	874.3587
NUMBER OF STAGES	---	NSTGBO	1.	NSTGBF	1.
SPECIFIC SPEED	---	SPBO	3999.9835	SPBF	3999.9735
EFFICIENCY	PERCENT	ETABO	66.2661	ETABF	71.2954
SHAFT SPECIFIC POWER	HP SEC/LB	SSHPBO	.6567	SSHPBF	2.2298
SHAFT POWER	HP	SHPBO	30.2203	SHPBF	17.1013
SHAFT POWER - RECIRCULATION FLOW	HP	HPRCBO	.0000	HPRCBF	.0000
FLOW COEFFICIENT-TIP-IMPELLER SUCTION	---	QCOSBO	.0734	QCOSBF	.0759
BLADE ANGLE, IMPELLER EXIT	DEG	BET2BO	7.0796	BET2BF	10.8413
RATIO HEAD COEFFICIENTS-ACTUAL TO TYPICAL	PERCENT	RHCMBO	89.3963	RHCMBF	111.8291
HEAD COEFFICIENT	---	HCOMBO	.3188	HCOMBF	.3988
FLOW COEFFICIENT	---	QCOMBO	.1236	QCOMBF	.1904
IMPELLER DIAMATER SUCTION - TIP	INCH	DTSBO	3.1346	DTSBF	3.6045
IMPELLER DIAMATER SUCTION - MEAN	INCH	DMSBO	2.3873	DMSBF	2.6610
IMPELLER DIAMATER SUCTION - HUB	INCH	DHSBO	1.2539	DHSBF	1.0814
IMPELLER DIAMATER EXIT - TIP	INCH	DTEBO	3.1346	DTEBF	3.3017
DIAMATER IMPELLER EXIT - MEAN	INCH	DMEBO	2.6260	DMEBF	2.8047
DIAMATER IMPELLER EXIT - HUB	INCH	DHEBO	1.9914	DHEBF	2.1980
IMPELLER MEAN SPEED - EXIT	FT/SEC	UPMBO	163.6173	UPMBF	279.9416
DENSITY PROPELLANT	LB/FT3	RHOBO	71.1686	RHOBF	4.3491
SPECIFIC HEAT, PROPELLANT	BTU/LB*DEG	CPBO	.3907	CPBF	2.8573
BOOST PUMP DRIVE					
HYDRAULIC TURBINE DRIVE		OXIDIZER TPA		FUEL TPA	
SHAFT POWER	HP	SHPTBO	30.2203	SHPTBF	17.1013
SHAFT SPEED, TURBINE	RPM	RPMTBO	14285.6532	RPMTBF	22856.9893

TABLE LI (cont.)

INLET PRESSURE	PSIA	PTIBO	769.8074	PTIBF	187.2154
EXIT PRESSURE	PSIA	PTEBO	535.6566	PTEBF	138.8261
PRESSURE RATIO TURBINE, IN/OUT	---	PRTBO	1.4371	PRTBF	1.3486
INLET TEMPERATURE	DEG R	TTIBO	165.5301	TTIBF	39.0192
EXIT TEMPERATURE	DEG R	TTEBO	165.9049	TTEBF	39.2072
IDEAL HEAD	FEET	DHTBO	473.7724	DHTBF	1602.1855
NUMBER OF STAGES	---	NTSTBO	1.	NTSTBF	1.
TURBINE NOZZLE ANGLE	DEGREE	ALPHBO	15.0000	ALPHBF	15.0000
BLADE/GAS VELOCITY RATIO	---	UCTBO	.5916	UCTBF	.6000
EFFICIENCY	PERCENT	ETATBO	76.2389	ETATBF	76.5447
SHAFT SPECIFIC POWER	HP SEC/LB	SSHTBO	.6567	SSHPTBF	2.2298
WEIGHT FLOW	LB/SEC	WGTBO	46.0166	WGTBF	7.6694
RATIO - TURBINE / PUMP FLOW	---	RTPBO	.0000	RTPBF	.0000
ROTOR MEAN SPEED	FT/SEC	UTMBO	105.9207	UTMBF	203.1152
ROTOR MEAN DIAMATER	INCH	DMTRBO	1.6993	DMTRBF	2.0366
ROTOR HUB DIAMATER	INCH	DHTRBO	1.4119	DHTRBF	1.6961
ROTOR TIP DIAMATER	INCH	DTTRBO	1.9656	DTTRBF	2.3524
HUB RATIO TURBINE ROTOR	---	HRTBO	.7261	HRTBF	.7286
NOZZLE ADMISSION	PERCENT	PCADBO	100.0000	PCADBF	100.0000
STRESS, CENTRIFUGAL FORCE, BLADE ROOT	PSI	SCFBBO	390.9092	SCFBBF	1422.2866
STRESS, AVERAGE TANGENTIAL, DISK	PSI	ATSDBO	370.2335	ATSDBF	1361.4402
SPECIFIC HEAT, TURBINE DRIVE FLUID	BTU/LB*DEG	CPTGBO	.3860	CPTGBF	2.5680

BOOST PUMP ASSEMBLY		OXIDIZER TPA		FUEL TPA	
DIAMATER, SHAFT, BEARING BORE	INCH	DSHFBO	.5925	DSHFBF	.5919
SHAFT TORSION, STRESS AT BEARING BORE	LB/IN2	SSHFBO	3266.2164	SSHFBF	1285.0323
POLAR MOMENT OF INERTIA	IN LB(SEC)2	ASYIBO	.0008	ASYIBF	.0011
MINIMUM START TIME	SEC	STBO	.0092	STBF	.0541

INDUCER - HIGH SPEED		OXIDIZER TPA		FUEL TPA	
WEIGHT FLOW	LB/SEC	WIO	46.0166	WIF	7.6694
VOLUME FLOW	GPM	QIO	290.3173	QIF	791.7912
PROPELLANT VAPOR PRESSURE	PSIA	PVIO	15.0000	PVIF	15.0000
PRESSURE SUCTION	PSIA	PSIO	141.2007	PSIF	43.2179
PRESSURE DISCHARGE	PSIA	PDIO	769.8074	PDIF	187.2154
TEMPERATURE SUCTION	DEG R	TSIO	163.4007	TSIF	38.1583
TEMPERATURE DISCHARGE	DEG R	TDIO	165.5301	TDIF	39.0192

CONTINEOUS RUN MINIMUM SUCTION CONDITION

NET POS., SUCTION PRESSURE, MIN CONT. OPER.	PSID	DPSRIO	126.2007	DPSRIF	28.2179
NPSH (CONT. MIN.) ABOVE VAPOR PRESSURE	FEET	HSVPIO	255.3500	HSVPIF	934.3037
THERMODYNAMIC SUPPRESION HEAD	FEET	TSHIO	4.0000	TSHIF	130.7936
NET POS. SUC. HEAD, CONT. MIN. EFFECTIVE	FEET	HSVIBO	259.3500	HSVNIF	1065.0972
SUCTION SPECIFIC SPEED, CONT OPER WITH THS	---	SSPNIO	13182.3167	SSPNIF	12074.0592
SUCTION SPECIFIC SPEED CONT. MIN. NO THS	---	SPRXIO	13336.8899	SPRXIF	13320.7683
SHAFT SPEED, PUMP	RPM	RPMIO	50000.0000	RPMIF	80000.0000

BREAKDOWN MINIMUM SUCTION CONDITION

RATIO RUN TO BREAKDOWN MIN EFFECTIVE NPSH	---	SFACIO	2.9935	SFACIF	3.3653
NET POSITIVE SUCTION PRESSURE, MINIMUM	PSID	DPSIO	40.8424	DPSIF	5.6085
NET POS. SUC. HEAD, MIN. ABOVE VAPOR PRES.	FEET	HSVMIO	82.6390	HSVMIF	185.7003
THERMODYNAMIC SUPPRESION HEAD	FEET	TSHIO	4.0000	TSHIF	130.7936
NET POSITIVE SUCTION HEAD, MIN. EFFECTIVE	FEET	HSVIO	86.6390	HSVIF	316.4939
SUCTION SPECIFIC SPEED, WATER EQUIVALENT	---	SSPIO	30000.0000	SSPIF	30000.0000
SUCTION SPECIFIC SPEED, BREAKDOWN, NO THS	---	SSPXIO	31082.6347	SSPXIF	44749.2534
PRESSURE RISE	PSID	DPIO	628.6067	DPIF	143.9975
HEAD RISE	FEET	DHIO	1271.9004	DHIF	4767.7995
NUMBER OF STAGES	---	NSTGIO	1.	NSTGIF	1.
SPECIFIC SPEED	---	SPIO	4000.0611	SPIF	3923.3480
EFFICIENCY	PERCENT	ETAIO	66.2660	ETAIF	71.3518
SHAFT SPECIFIC POWER	HP SEC/LB	SSHPIO	3.4898	SSHPIF	12.1493
SHAFT POWER	HP	SHPIO	160.5885	SHPIF	93.1782
SHAFT POWER - RECIRCULATION FLOW	HP	HPRCIO	.0000	HPRCIF	.0000
FLOW COEFFICIENT-TIP-IMPELLER SUCTION	---	QCOSIO	.1000	QCOSIF	.0997
BLADE ANGLE, IMPELLER EXIT	DEG	BET2IO	10.1861	BET2IF	10.1880
RATIO HEAD COEFFICIENTS-ACTUAL TO TYPICAL	PERCENT	RHCMIO	91.9445	RHCMIF	91.5470
HEAD COEFFICIENT	---	HCOMIO	.2973	HCOMIF	.3030
FLOW COEFFICIENT	---	QCOMIO	.1789	QCOMIF	.1789
IMPELLER DIAMATER SUCTION - TIP	INCH	DTSIO	1.9446	DTSIF	2.3278
IMPELLER DIAMATER SUCTION - MEAN	INCH	DMSIO	1.5448	DMSIF	1.8515
IMPELLER DIAMATER SUCTION - HUB	INCH	DHSIO	.9956	DHSIF	1.1988
IMPELLER DIAMATER EXIT - TIP	INCH	DTEIO	1.9446	DTEIF	2.3278
DIAMATER IMPELLER EXIT - MEAN	INCH	DMEIO	1.6993	DMEIF	2.0366
DIAMATER IMPELLER EXIT - HUB	INCH	DHEIO	1.4119	DHEIF	1.6961
IMPELLER HEAD SPEED - EXIT	FT/SEC	UPMIO	371.0208	UPMIF	711.4769
DENSITY PROPELLANT	LB/FT3	RHOIO	71.1686	RHOIF	4.3491
SPECIFIC HEAT, PROPELLANT	BTU/LB*DEG	CPIO	.3907	CPIF	2.8573

TURBOPUMPS

PUMPS - CENTRIFUGAL		OXIDIZER TPA		FUEL TPA	
WEIGHT FLOW	LB/SEC	WMO	46.0166	WMF	7.6694
VOLUME FLOW	GPM	QO	290.3173	QF	791.7912
PROPELLANT VAPOR PRESSURE	PSIA	PVMO	15.0000	PVMF	15.0000
PRESSURE SUCTION	PSIA	POS	535.6566	PFS	138.8261
PRESSURE DISCHARGE	PSIA	POD	3092.4000	PFD	4282.7388
TEMPERATURE SUCTION	DEG R	TSMO	165.9049	TSMF	39.2072
TEMPERATURE DISCHARGE	DEG R	TOD	175.5620	TFD	77.7544
CONTINEOUS RUN MINIMUM SUCTION CONDITION					
NET POS., SUCTION PRESSURE, MIN CONT. OPER.	PSID	DPSRMO	520.6566	DPSRMF	123.8261
NPSH (CONT. MIN.) ABOVE VAPOR PRESSURE	FEET	HSVORP	1053.4780	HSVFRP	4099.9178
NET POS. SUC. HEAD, CONT. MI., EFFECTIVE	FEET	HSVORN	1057.4780	HSVFRN	4241.1371
SUCTION SPECIFIC SPEED, CONT OPER WITH THS	---	SSPORN	4594.1260	SSPFRN	4283.3496

TABLE LI (cont.)

SUCTION SPECIFIC SPEED CONT. MIN. NO THS	---	SSPORX	4607.2032	SSPFRX	4393.5338
SHAFT SPEED, PUMP	RPM	RPMO	50000.0000	RPMF	80000.0000
BREAKDOWN MINIMUM SUCTION CONDITION					
RATIO RUN TO BREAKDOWN MIN EFFECTIVE NPSH	---	SFACTO	2.8210	SFACTF	3.0971
NET POSITIVE SUCTION PRESSURE, MINIMUM	PSID	DPOS	183.2914	DPFS	37.0933
NET POS. SUC. HEAD, MIN. ABOVE VAPOR PRES.	FEET	HSVOVP	370.8653	HSVFMP	1228.1701
THERMODYNAMIC SUPPRESION HEAD	FEET	TSHO	4.0000	TSHF	141.2195
NET POSITIVE SUCTION HEAD, MIN., EFFECTIVE	FEET	HSVO	374.8653	HSVF	1369.3896
SUCTION SPECIFIC SPEED, WATER EQUIVALENT	---	SSPO	10000.0000	SSPF	10000.0000
SUCTION SPECIFIC SPEED, BREAKDOWN, NO THS	---	SSPOX	10080.7842	SSPFX	10850.5415
PRESSURE RISE	PSID	DPO	2556.7434	DPF	4143.9127
HEAD RISE	FEET	DHO	5173.2230	DHF	137206.1875
NUMBER OF STAGES	---	NSTGO	1.	NSTGF	3.
SPECIFIC SPEED	---	SPO	1396.6434	SPF	719.7903
EFFICIENCY	PERCENT	ETAO	63.7912	ETAF	61.5483
SHAFT SPECIFIC POWER	HP SEC/LB	SSHPO	14.7448	SSHPF	405.3173
SHAFT POWER	HP	SHPO	678.5044	SHPF	3108.5573
SHAFT POWER - RECIRCULATION FLOW	HP	HPRCRO	33.9252	HPRCRF	217.5990
BLADE ANGLE, IMPELLER EXIT	DEG	BET20	23.0130	BET2F	73.2608
HEAD COEFFICIENT	---	HCOEFO	.4766	HCOEFF	.6502
FLOW COEFFICIENT	---	QCOEFO	.1192	QCOEFF	.0982
IMPELLER DIAMATER SUCTION - TIP	INCH	DTSMO	1.9446	DTSMF	2.3278
IMPELLER DIAMATER SUCTION - MEAN	INCH	DMSMO	1.6993	DMSMF	2.0366
IMPELLER DIAMATER SUCTION - HUB	INCH	DHSMO	1.4119	DHSMF	1.6961
IMPELLER DIAMATER EXIT - TIF	INCH	DTEMO	2.7076	DTEMF	4.3080
DIAMATER IMPELLER EXIT - MEAN	INCH	DMEMO	2.7076	DMEMF	4.3080
DIAMATER IMPELLER EXIT - HUB	INCH	DHEMO	2.7076	DHEMF	4.3080
IMPELLER TIP SPEED	FT/SEC	UPTMO	591.1682	UPTMF	1504.9922
IMPELLER EXIT BLADE HEIGHT	INCH	HIBO	.1548	HIBF	.1266
DISCHARGE PONT DIAMATER	INCH	DDPO	.7249	DDPF	.4736
WEIGHT FLOW, HALF STAGE	LB/SEC	WO2	6.6930	WF2	.0000
VOLUME FLOW, HALF STAGE	GPM	QO2	42.2257	QF2	.0000
PRESSURE SUCTION, HALF STAGE	PSIA	POS2	2799.1771	PFS2	.0000
PRESSURE DISCHARGE, HALF STAGE	PSIA	POD2	4440.9625	PFD2	.0000
TEMPERATURE SUCTION, HALF STAGE	DEG R	TOS2	175.5620	TFS2	.0000
TEMPERATURE DISCHARGE, HALF STAGE	DEG R	TOD2	187.9561	TFD2	.0000
PRESSURE RISE, HALF STAGE	PSID	DPO2	1641.7854	DPF2	.0000
HEAD RISE, HALF STAGE	FEET	DHO2	3321.9298	DHF2	.0000
SPECIFIC SPEED, HALF STAGE	---	SPO2	742.5326	SPF2	.0000
EFFICIENCY, HALF STAGE	PERCENT	ETAO2	46.8500	ETAF2	.0000
SHAFT SPECIFIC POWER, HALF STAGE	HP SEC/LB	SSHPO2	12.8919	SSHPF2	.0000
SHAFT POWER, HALF STAGE	HP	SHPO2	86.2856	SHPF2	.0000
BLADE ANGLE, IMPELLER EXIT, HALF STAGE	DEG	BET2O2	90.0000	BET2F2	.0000
HEAD COEFFICIENT, HALF STAGE	---	HCOEO2	.5500	HCOEF2	.0000
FLOW COEFFICIENT, HALF STAGE	---	QCOEO2	.0350	QCOEF2	.0000
DENSITY PROPELLANT	LB/FT3	RHOMO	71.1686	RHOMF	4.3491
SPECIFIC HEAT, PROPELLANT	BTU/LB*DEG	CPMO	.3907	CPMF	2.8573

TURBINES - AXIAL FLOW	OXIDIZER TPA	FUEL TPA

TABLE LI (cont.)

SHAFT POWER	HP	SHPTO	959.3037	SHPTF	3419.3343
SHAFT SPEED, TURBINE	RPM	RPMTO	50000.0000	RPMTF	80000.0000
INLET PRESSURE	PSIA	PTIO	2972.3753	PTIF	2972.3753
EXIT PRESSURE	PSIA	PTEO	1994.9400	PTEF	1994.9400
PRESSURE RATIO TURBINE, IN/OUT	---	PRTO	1.4900	PRTF	1.4900
INLET TEMPERATURE	DEG R	TTIO	1860.0000	TTIF	1860.0000
EXIT TEMPERATURE	DEG R	TTEO	1773.6245	TTEF	1740.4983
IDEAL HEAD	FEET	HTO	288893.6758	HTF	288893.6758
NUMBER OF STAGES	---	NTSTGO	1.	NTSTGF	2.
TURBINE NOZZLE ANGLE	DEGREE	ALPHAO	15.0000	ALPHAF	20.0000
BLADE/GAS VELOCITY RATIO - PER STAGE	---	UCTO	.2550	UCTF	.4262
EFFICIENCY	PERCENT	ETATO	47.3718	ETATF	65.5395
SHAFT SPECIFIC POWER	HP SEC/LB	SSHPTO	248.8256	SSHPTF	344.2535
WEIGHT FLOW	LB/SEC	WGTO	3.8553	WGTF	9.9326
RATIO - TURBINE / PUMP FLOW	---	RTPO	.0838	RTPF	1.2951
ROTOR MEAN SPEED	FT/SEC	UTMO	1100.0000	UTMF	1300.0000
ROTOR MEAN DIAMATER	INCH	DMTRO	5.0380	DMTRF	3.7212
DIAMATER ROTOR HUB, FIRST STAGE	INCH	DHTRO1	4.7728	DHTRF1	3.4959
DIAMATER ROTOR, FIRST STAGE	INCH	DTTRO1	5.3032	DTTRF1	3.9466
HUB RATIO TURBINE ROTOR, FIRST STAGE	---	HRTO1	.9000	HRTF1	.8858
ROTOR MEAN DIAMATER	INCH	DMTRO	5.0380	DMTRF	3.7212
ROTOR HUB DIAMATER,LAST STAGE	INCH	DHTRO	4.7728	DHTRF	3.4215
ROTOR TIP DIAMATER,LAST STAGE	INCH	DTTRO	5.3032	DTTRF	4.0210
HUB RATIO TURBINE ROTOR, LAST STAGE	---	HRTO	.9000	HRTF	.8509
NOZZLE ADMISSION	PERCENT	PCADMO	31.2464	PCADMF	100.0000
STRESS, CENTRIFUGAL FORCE, BLADE ROOT	PSI	SCFBMO	14146.1681	SCFBMF	30238.0427
STRESS, AVERAGE TANGENTIAL, DISK	PSI	ATSDMO	39929.9995	ATSDMF	55769.9995
TURBINE NOZZLE FLOW PARAMATER (W*T**.5/(A*P)--		FLOPMO	.1780	FLOPMF	.1468
AREA - TURBINE NOZZLE	IN2	ATURMO	.3142	ATURMF	.9818
SPECIFIC HEAT, TURBINE DRIVE GAS	BTU/LB*DEG	CPTGO	2.0091	CPTGF	2.0091
SPECIFIC HEAT RATIO, TURBINE DRIVE GAS	---	GAMTO	1.3491	GAMTF	1.3491
GAS CONSTANT, TURBINE DRIVE GAS	FT/DEG R	RTO	409.9572	RTF	409.9572
TURBOPUMPS - INLINE		OXIDIZER TPA		FUEL TPA	
DIAMATER,SHAFT, BEARING BORE	INCH	DSHFTO	1.1811	DSHFTF	.9843
SHAFT TORSION STRESS AT BEARING BORE	LB/IN2	SSHFTO	3740.1045	SSHFTF	14397.6937
POLAR MOMENT OF INERTIA	IN LB(SEC)2	OASYI	.0078	FASYI	.0148
MINIMUM START TIME	SEC	STO	.0337	STF	.0458

minimum. For 16 feet NPSH without a boost pump, a turbopump would operate at 14,300 rpm and weigh in excess of 300 lb. The oxidizer turbopump with a 4.5 lb boost turbopump can operate at the desired 50,000 rpm and weigh 25.6 lb giving a combined weight of 30.1 lb.

A low speed boost pump is utilized to supply the differential pressure between the specified minimum and the minimum required by the high speed pump. The boost pump concept selection and its design parameters are discussed below under low speed inducer.

The turbopump power balance includes allocated internal flows for LO_2 cooling of the pump end bearing, LH_2 cooling of the turbine end bearing, and LH_2 leakage from the turbine end bearing into the turbine exhaust.

The allocated flows, the predicted value for the designs, and the flow control means are presented below:

LO_2 TURBOPUMP INTERNAL RECIRCULATION FLOW

Flow Location	Predicted lb/sec	Predicted Percent Pump Flow	Seal Type	Percent of Total Oxid. Flow	Allocated Percent
First Stage Impeller Front Side	.695	1.50	Hydrostatic Face Seal	4.55	7% in first stage
First Stage Impeller Thrust Brg. Flow	.40	.88	Hydrostatic Shaft Seal		
Thrust Balancer	1.0	2.17	Hydrostatic		
Second Stage Front Side	.38	.83	Hydrostatic Face Seal	2.85	Included in first stage allocation.
Second Stage Return to 2nd Stage Suction	.66	1.43	Hydrostatic Face Seal		
Second Stage Overboard	.27	.59	Labyrinth		
Helium Purge Seal	.0045	----	Static Lift-Off Seal	----	----

LO_2 TURBOPUMP INTERNAL RECIRCULATION FLOW (cont.)

Flow Location	Predicted lb/sec	Predicted Percent Pump Flow	Seal Type	Percent of Total Oxid. Flow	Allocated Percent
Fuel Overboard Seal	.012	.16*	Labyrinth	.91* (Fuel Overboard Seal and Fuel Return to Fuel TPA combined)	Included in LH_2 pump recirculation allocation.
Fuel Return to Fuel TPA	.07	.91*	Labyrinth		
Fuel through Turbine Seal to Turbine Exhaust	.188	8.7**	Labyrinth	----	4% ** Bypassing Turbine

* Percent of LH_2 Pump Flow
** Percent of LH_2 in Turbine Flow

The effective power consumption from recirculation flow of the combined main and half-stage is 4.55% for the main plus an effective power consumption of 1.51% from the second stage giving a total of 6.06%. The allocated recirculation flow power consumption from the pump system is 7.0%. The 1.51% effective power consumption from the second stage recirculation is derived from:

$$\text{Percent Power} = \text{Percent Recirculation} \times \frac{\Delta P \text{ Half-Stage}}{\Delta P \text{ Boost} + \text{Main Stage}}$$

$$= 2.85\% \times \frac{1625 \text{ psi}}{3077 \text{ psi}}$$

$$= 1.51\%$$

The overboard LO_2 and LH_2 flows will be mixed with GHe purge flow. These flows can be dumped overboard or they can be used towards the pressurizing of the propellant tanks.

The turbopump power balance computations were performed with an allocated 4% of the LH_2 going to the oxidizer TPA turbine flowing into the turbine exhaust. Analysis of the TPA design showed that this flow would be 8.7%.

The impact of the predicted value exceeding the allocated value will have little effect on the power balance. The calculated total LH_2 leakage flow into the turbine exhaust for the fuel and oxidizer turbopump turbines is 0.188 + 0.188 = 0.376 lb/sec which is 4.9%

of the LH_2 flow. The allocated leakage flow into the turbine is 4% or 3.07 lb/sec. This slight excess of the predicted over the allocated value is noted, but was considered to be of sufficiently small magnitude that all parametric and design point power balance computations for this study will be based on the 4% allocation value.

<u>2</u> Main Oxidizer Pump

The main pump includes the high speed inducer and a main and a half-stage to develop the required pressures at engine rated thrust. In addition to developing the required pressure at the rated operating point, the pump is required to operate at flow coefficient values as low as 45% of design where the engine is throttled to 20% of rated thrust. Inducer suction performance is degraded for this reduced flow coefficient operation as shown in Figure 195 for the high speed inducer and the suction of the centrifugal main stage. The design approach for the high speed inducer and the inlet of the high speed main are the same as described for the LH_2 pump above. The high speed inducer S/S_{design} value is 0.38, thus, the maximum operating suction specific speed is 30,000 x 0.38 = 11,400 and the ratio of RUN-to-BREAKDOWN NPSH value is 2.99. The high speed centrifugal impeller S/S_{design} value is 0.43, and the maximum operating suction specific speed is 10,000 x 0.43 = 4300 and the ratio of RUN-to-BREAK-DOWN NPSH value is 2.82. The ratio of RUN-to-BREAKDOWN values for the high speed inducer and the suction of the centrifugal main exceed the design minimum value of 1.87 for the same reason as described for the fuel pumps above.

<u>3</u> Main Turbine

The design objectives of oxidizer TPA turbine include the requirements as described for the fuel TPA turbine plus the added requirement that the TPA start spin-up time be of the same order of magnitude as the fuel TPA. The turbine contributes the majority of the polar moment of inertia and the turbine diameter and number of stages defines the polar moment of inertia. Therefore, a compromise to achieving maximum efficiency is dictated by this spin-up rate requirement.

A radial inflow and an axial flow turbine were considered for this application. The axial flow design was chosen because of its higher efficiency. One-, two- and three-stage turbines were considered and both velocity and pressure staging was investigated. A turbine with a single stage, 1100 ft/sec mean blade speed, and a partial admission nozzle design was selected for the oxidizer TPA since: (1) it is the simplest design mechanically, (2) results in an acceptable critical speed operation of the TPA, and (3) delivers acceptable efficiency. The turbopump power balance was computed to an allocated turbine efficiency value of 47.5%; this value based on a single-stage partial admission configuration. The turbine has a predicted efficiency of 52% which meets and exceeds the allocated 47.5%. The turbine efficiency could be increased two points with a velocity compounded two-stage configuration; however, a two-stage turbine

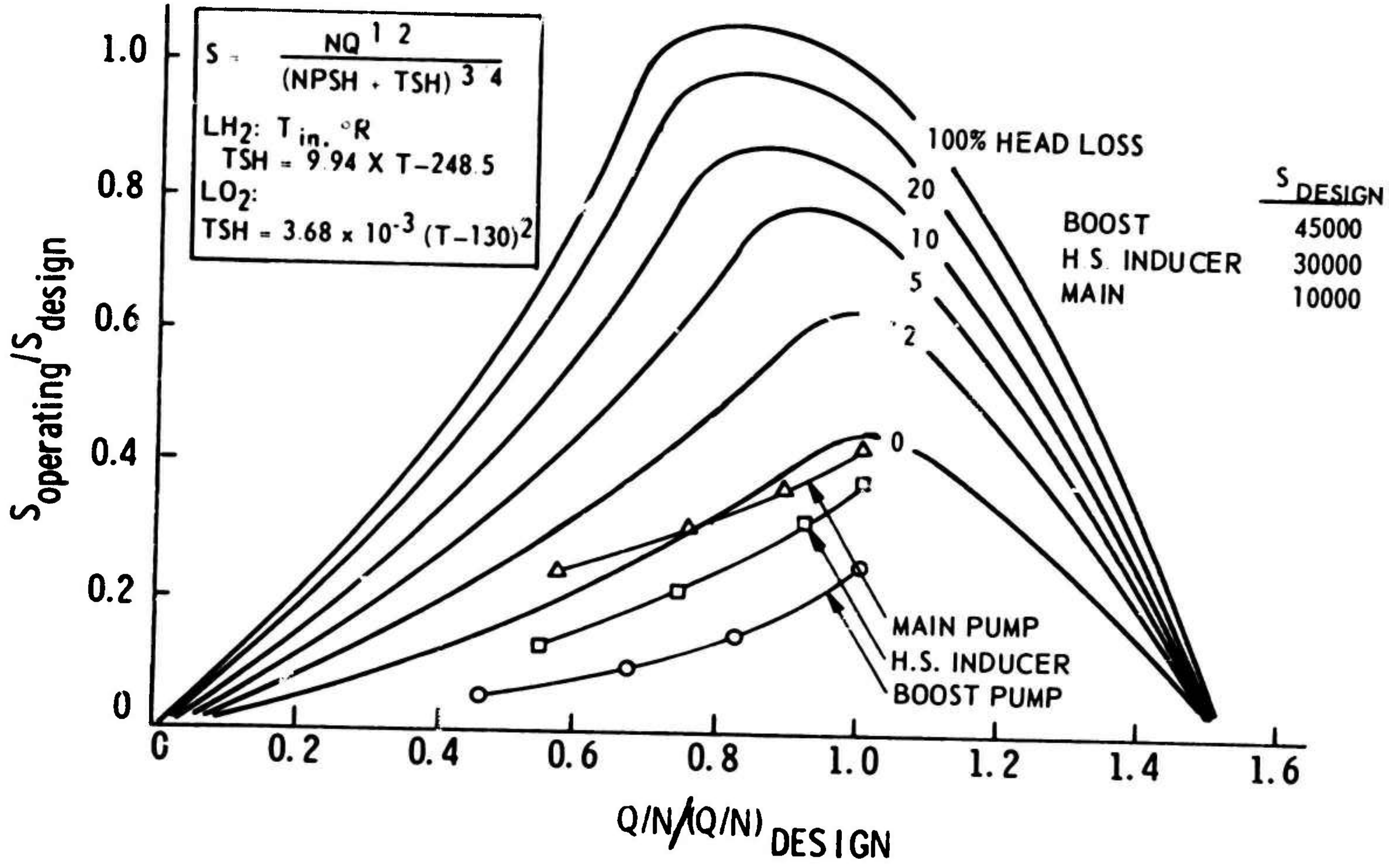


Figure 195. Normalized Oxidizer Pump Headloss vs Suction Specific Speed and Q/N

configuration did not result in an acceptable TPA critical speed operating region to accommodate engine throttling. A pressure compounded two-stage turbine was not considered due to the low turbine gas flow rate which results in a partial rather than a full nozzle admission turbine.

A turbine mean blade speed of 1100 ft/sec was selected so that the start times of oxidizer TPA would be of the same order as fuel TPA.

4 Power Transmission

The power transmission portion of the high speed oxidizer turbopump is required to support the rotating elements of the turbopump assembly, control internal flow of LO_2 and LH_2, and provide a safe separation of LO_2 and LH_2. The allocated and calculated values of internal flows and the means for their control are discussed above under LO_2 TPA Design Parameters.

The shaft breakaway torque shall not exceed 6 in.-lb which is 50% of the turbine stall torque developed from a tank pressure start with combustion in the primary combustor. Preloaded duplex ball bearings and static lift-off seals were selected to achieve a minimum breakaway torque value.

The bearings have been designed to have a 0.2 B_1 life of 10 hr. This was achieved by limiting the maximum axial thrust load of the life-limited bearing to 78 lb and the pump radial load to 100 lb (Figure 196). This 78 lb is the summation of 25 lb preload and a 53 lb applied load. The 0.2 factor to the B_1 bearing life accounts for the life reduction resulting from cryogenic operation. Refer to the Main Fuel Turbopump - Fuel Power Transmission section of this report.

(c) Low Speed Pumps

In order to obtain a minimum number of pump and turbine stages and an acceptable weight for the main turbopumps, low speed inducers are required upstream of the main pumps. The inducer speed, size, and design characteristics are readily established and straight forward: however, the selection of the inducer drive from candidate systems (Figure 197) involves trade-off studies with respect to both turbomachinery and engine system and design characteristics.

1 Low Speed Pump Drive Selection

Six different types of low speed inducer drives were evaluated. These include full flow hydraulic turbine, partial-admission hydraulic tip turbine, multi-stage hub hydraulic turbine, hydrogen gas turbine, mechanical speed reducing drives, and electric motor drive. The full flow hydraulic drive turbine was selected for the fuel and oxidizer TPA (Figures 188 and 194) because: (1) it does not introduce two-phase flow to

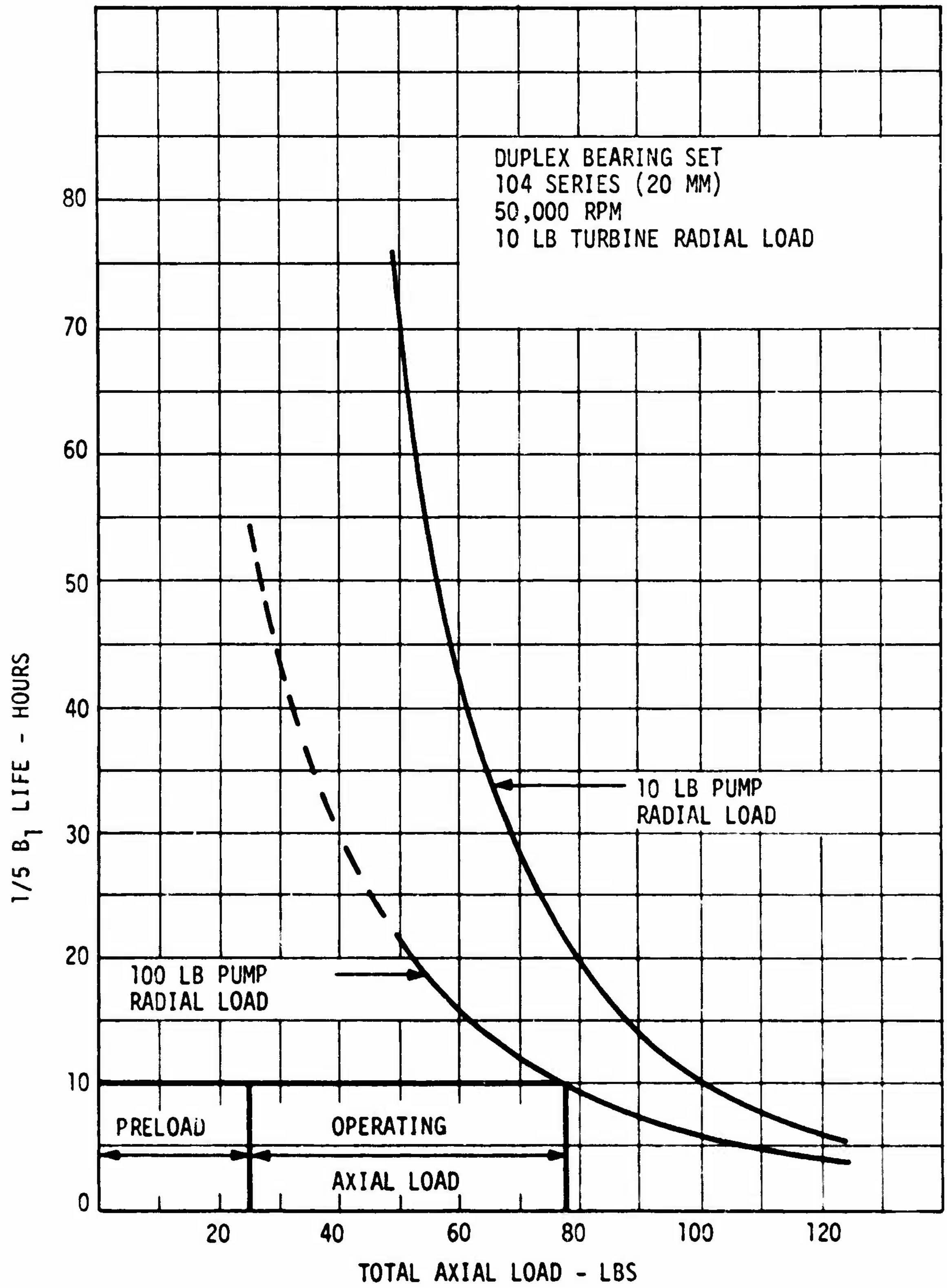


Figure 196. Oxidizer TPA Turbine End Duplex Bearing Set Life vs Axial Load

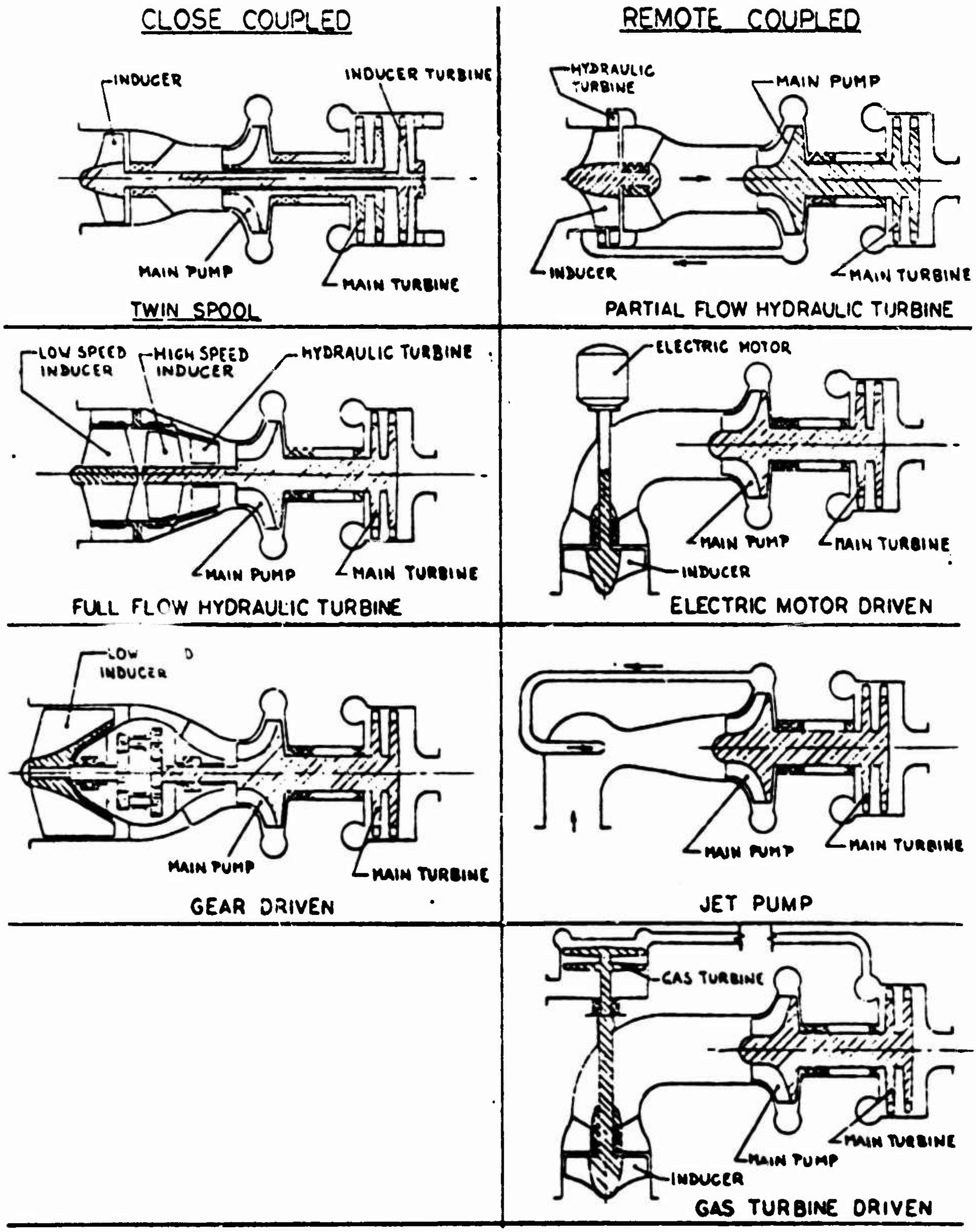


Figure 197. Boost Pump Types

the high speed inducer, (2) it has high efficiency which reduces required fuel pump discharge pressure by 150 psi and, (3) the low speed inducer can be integral with the high speed pump. Table LII gives the significant advantages and disadvantages of inducer drive options. The tip turbine drive is an attractive alternate; however, it requires approximately 10 to 20 diameters of line-length between the low and high speed inducer to accomplish the mixing of the flow leaving the tip turbine and the low speed inducer flow. This mixing of turbine exhaust flow which is part liquid plus vapor with the main flow stream at the exit of the low speed inducer requires the vapor to be cooled and condensed prior to its entry into the high speed inducer.

2 Boost Pump Design Parameters

The design parameters for the LH_2 and LO_2 low speed boost pumps are given in Table XLIX. These boost turbopumps include the low speed inducers, full flow hydraulic turbines, and the power transmission sections. The boost pumps are required to supply propellant flow to the high speed inducers over a range of operating conditions from rated engine thrust to 20% of engine rated thrust. Throttling the engine over this range results in the requirement that the boost pump operate from its design flow coefficient at engine rated thrust down to 46% of design flow coefficient at 20% engine rated thrust. The predicted head loss over the flow coefficient range expressed as: $(Q/N)/(Q/N)_{Design}$, is given in Figures 191 and 195 for the fuel and oxidizer boost pumps, respectively.

The design approach to the low speed inducers is the same as for the high speed inducers discussed above under the heading High Speed Main Fuel Turbopump. An additional constraint is added to the boost pump design, this being a limitation on the maximum boost pump shaft speed relative to the high speed main pump shaft speed. A ratio of shaft speeds, high speed main-to-low speed boost pump, was assigned a minimum value of 3 to 5 for this study. Designing the propellant feed system with the main pump shaft speed value approaching the shaft speed value of the boost pump suggest the elimination of the low speed boost pump and designing the main pump to operate at a reduced speed of the boost pump. The design shaft speed for the fuel and oxidizer low speed boost pumps were established by this limiting shaft speed ratio.

The full flow hydraulic turbine is driven by the propellant as it leaves the high speed inducer. The energy available to drive this turbine is the swirl velocity leaving the high-speed inducer.

TABLE LII

LOW SPEED PUMP DRIVE EVALUATION

FULL FLOW HYDRAULIC DRIVE	PARTIAL FLOW HYDRAULIC DRIVE	MULTISTAGE HUB TURBINE DRIVE	ELECTRICAL MOTOR DRIVE	HOT GAS TURBINE DRIVE
ADVANTAGES • HIGH EFFICIENCY • NO DYNAMIC SEALS • NO TWO-PHASE FLOW IN TURBINE • NO EXTERNAL HIGH PRESSURE LINES	ADVANTAGES • CAN BE MOUNTED ON VEHICLE • SIMPLE DESIGN • COMPACT DESIGN ON CLOSE COUPLED • LOW PRESSURE DIFFERENTIAL DYNAMIC SEALS	ADVANTAGES • DYNAMIC SEALS BETWEEN PUMP AND TURBINE AT SMALLER DIA THAN TIP TURBINE • TURBINE CLEARANCE MORE READILY MAINTAINED THAN TIP TURBINE	ADVANTAGES • CAN BE MOUNTED ON VEHICLE • CAN BE USED FOR CHILLDOWN • BOOST PUMP CAN BE STARTED FIRST DURING START TRANSIENT	ADVANTAGES • CAN BE MOUNTED ON VEHICLE • MORE CONTROL OVER START TRANSIENT • NO ADDITIONAL INEFFICIENCIES DUE TO HYDRAULIC TURBINE OR MOTOR/GENERATOR
DISADVANTAGES • BOOST PUMP FOLLOWS MAIN PUMP DURING START TRANSIENT • LONG TURBOPUMP ASSEMBLIES • COMPLEX MECHANICAL DESIGN • CANNOT BE LOCATED REMOTE FROM MAIN PUMP	DISADVANTAGES • HIGH PRESSURE LINE ACROSS GIMBAL WHEN VEHICLE MOUNTED • BOOST PUMP FOLLOWS MAIN PUMP DURING START TRANSIENT • HYDRAULIC TURBINE DISCHARGE IS TWO PHASE • NOT AS EFFICIENT AS FULL FLOW	DISADVANTAGES • HIGH TURBINE AXIAL THRUST • LARGE NUMBER OF STAGES TO OBTAIN EFFICIENCY (POSSIBLY 4) • COMPLEX MULTISTAGE DESIGN	DISADVANTAGES • REQUIRES BATTERY DURING START TO GAIN ADVANTAGES OF INDEPENDENT OPERATION • REQUIRES POWER TAKEOFF FOR GENERATOR • COULD BE HEAVY DEPENDING ON SIZE MOTOR	DISADVANTAGES • REQUIRES HIGH PRESSURE HOT GAS FLEX LINE ACROSS GIMBAL PLANE FOR VEHICLE MOUNTED • MORE HOT GAS LINES • HIGH PRESSURE HOT GAS SEAL • INTERPROPELLANT SEAL FOR OXIDIZER INDUCER • HIGH AXIAL THRUST

(3) Description of Turbopumps

(a) Fuel Turbopump

1 Assembly

Two conceptual design approaches have been taken for the main fuel turbopump, a three-stage pump configuration shown in Figure 198. The three-stage design which was selected as the baseline design includes a low speed boost pump with a full-flow turbine drive.

The three-stage pump configuration consists of a conventional front-to-back shrouded impeller pump arrangement driven by a two-stage pressure compounded turbine. The pump impellers, couplings and tension bolt form the shaft of the turbopump which is mounted on preloaded angular contact bearings. The bearings are propellant cooled and are located between the first- and second-stage impellers at the pump end, and inboard of the turbine rotors at the turbine end. A single-acting hydrostatic axial thrust balancer is located on the backside of the third-stage impeller. The bearings are mounted in retainers which contain preload springs and the retainers are spring-loaded radially to provide damping as the shaft passes through the "rigid body" critical speeds. The turbopump operates at 80,000 rpm at the nominal engine thrust and mixture ratio value of 25K and 6.0, respectively. The turbopump operating speed range is above the first- and second-critical speed, but below the third-critical speed (shaft bending). The turbine end bearing retainer is spring-loaded axially so that the bearings can operate satisfactorily with the axial thrust balancer. The pump end bearings are free to float axially. The turbine bearings take unbalanced axial thrust during the transients and the thrust balancer takes the unbalanced thrust during steady-state when pump pressure is available. During steady-state operation, the external axial load on the turbine bearings is from the retainer spring only (approximately 50 lb) and no external axial load is taken by the pump end bearings since they are free to float. The thrust balancer consists of a hydrostatic face seal which acts as the inlet flow restrictor, close-running stationary and rotating inboard which act as the exit flow restrictor, and a cavity between these restrictors. The balancer is an axial clearance (exit restrictor) sensitive device which compensates for unbalanced thrusts by allowing pressure changes in the cavity between restrictors. If an unbalanced thrust occurs on the shaft, this thrust causes the clearance of the exit restrictor to change causing the pressure in the cavity between restrictors to change (either increase or decrease), thereby providing a counteracting force for the initial unbalanced thrust. A static lift-off seal is located between the turbine and turbine end bearing. This seal provides static sealing of LH_2 when the turbopump is not operating and allows a controlled amount of leakage into the turbine during operation.

The low speed inducer and turbine shaft is located on its bearings and is decoupled mechanically from the high speed shaft.

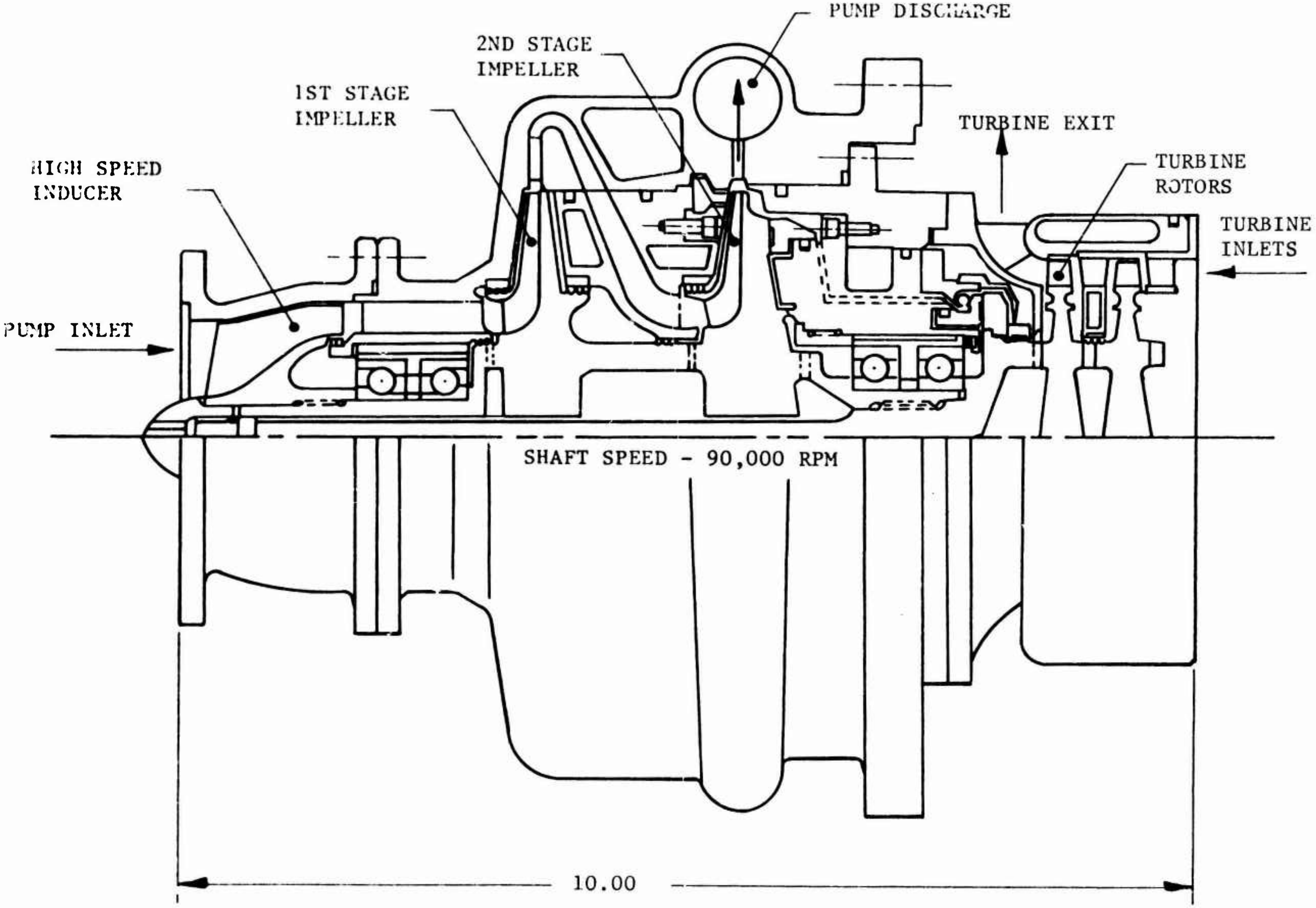


Figure 198. OOS Fuel Turbopump-2 Stage

2 Fuel Pump

The pump consists of shrouded impellers discharging into multiple-vaned housings with the first-stage impeller preceded by a high speed inducer. Vanes in the housings are provided to guide the flow, minimize radial thrust, and to act as structural members for containment of the high pressure. Hydrostatic seals on the impeller shrouds are placed at the diameters necessary to balance axial thrust and provide a bias load for the thrust balancer in the direction toward the turbine. These impeller seals and the shaft interstage hydrostatic seals operate at clearances of .0008-in. Leakage values through these seals are defined below under Design Selection - Fuel Power Transmission. This design approach results in a single-piece pump housing that does not require heavy high pressure flanges and static seals. Several pump configurations were considered and investigated before selecting this concept. Included in these configurations were open impeller designs, back-to-back impellers with a crossover housing, and reversed orientation impellers with a radial pump inlet. The selected configuration is a compact conventional pump package with flow paths as short as possible and a symmetrical housing which utilizes all members for either structural or hydraulic purposes.

3 Fuel Turbine

The fuel TPA turbine (Figure 188) has two stages that are pressure compounded with 100% nozzle admission and impulse blading. The turbine rotors are over-hung on the TPA shaft with the exhaust side of the turbine being adjacent to the turbine end bearings. Locating the turbine exhaust adjacent to the bearing cavity results in the turbine seal leakage flowing into the turbine exhaust stream where it will not reduce turbine efficiency. The turbine, which includes the nozzle assembly, attaches to the preburner housing by means of a bolt circle.

4 Power Transmission

a Rotor Support

Two important operational characteristics which exist for the rotor and bearing system are axial thrust and rotor system dynamics. Axial thrust loads in high pressure turbopumps can be several orders-of-magnitude higher than the bearing capacities. Thus, an axial thrust balancer is required in order to obtain a high enough TPA operating speed to achieve the required pump and turbine efficiencies (by virtue of their specific speeds) without using a prohibitive number of stages. The turbopump must operate above the "rigid body" critical speeds. This required unique approaches for the bearing support design to achieve the required effective bearing spring rate.

The resolution of axial forces in the fuel TPA has been accomplished by a system of pressurized areas, rolling element bearings (for transient rotor positioning) and a self-compensating

hydrostatic thrust balancer. The latter component is located on the backside of the last stage impeller. In this design, the first- and second-stage are thrust balanced by hydrostatic seals on both the front and backsides with the cavities inside the back seal vented to impeller inlets. The frontside pressure area force on the last-stage impeller resists the turbine shaft end thrust and provides the bias load for the single-acting thrust balancer. This thrust balancer has a maximum capacity of approximately 20,000 lb. Since the TPA operates above the first two critical speeds, this thrust balancer is designed to accommodate rocking and translational motions while transgressing the two critical speeds. Critical speed considerations dictate the rolling element bearing location with the turbine end bearing being a compromise between a desirable large shaft diameter for shaft stiffness and small shaft diameter for low DN. The pump end bearing location is between the first- and second-stage impeller to provide a reasonable bearing span.

b Fuel TPA Bearings

A preliminary analysis and selection of bearing sizes and configuration was made for the three- and two-stage turbopump conceptual designs. This analysis indicated that the same bearing sizes and arrangements could be used for the two TPA concepts. 105 size "DB" duplex bearings were selected for the turbine end and 104 size "DB" duplex bearings were selected for the pump end. The rotating speed (80,000 rpm) results in the following DN values:

Bearing Location	Bore Size	DN
Pump	20 mm	1.6×10^6 rpm x mm
Turbine	25 mm	2.0×10^6 rpm x mm

The 0.2 factored B_1 fatigue lives of the bearing preliminary designs appear adequate. The turbine end bearing lives are most sensitive to axial thrust load. This load shall not exceed 75 lb (25 lb preload plus 50 lb applied load) to achieve a 0.2 factored B_1 life of 10 hours. The radial loads shall be limited to 10 lb from the turbine and 100 lb from the pump. A life-reduction factor of 0.2 is applied to the B_1 life to account for the low viscosity coolant. The predicted 0.2 B_1 bearing lives for each of the four bearings starting from the pump end are 50.1, 49.6, 10.2, and 18.6 hours.

c Critical Speed

A preliminary critical speed analyses for the fuel turbopump was conducted for the TPA configuration shown in Figure 188. A rotor-only shaft critical speed model is applicable

for the turbopump because the bearings are "soft" mounted radially. The "soft" mounting is employed to tune the critical speeds such that the turbopump will operate above the rigid body criticals and below the shaft bending critical. Moreover, the mounting will be stiff enough to preclude large deflections due to pump-side loads and whirl forces.

The turbopump normal power level (NPL) speed is 80,000 rpm, and the minimum power level (MPL) speed is 35,000 rpm. Design criteria states that if operation is above the rigid body critical speed, the shaft bending critical speed shall be a minimum of 1.15 times the maximum operating speed and the rigid body critical speeds shall be no more than 0.85 times the minimum operating speed. The analysis for the fuel turbopump was accomplished using a computerized method for predicting bearing loads, shaft deflections, and critical speeds for shafts coupled by rolling contact bearings to the turbopump housing. The bearing nonlinearities, casing as well as rotor dynamics, and rotor-imbalance forcing function all can be included in the system dynamics analysis. Basically, the program has the capability for analyzing the forced-undamped, lateral vibrations of two elastically-coupled lumped-parameter beams. The program computes the amplitudes of the shears, moments, slopes, and deflection attributable to harmonic forcing functions. Shear deflections, rotary inertia, and gyroscopic effects for rotating shaft analyses are also included.

The results of the rotor-only parametric analysis are shown in Appendix A where the first three rotor critical speeds are plotted versus effective radial stiffness at the bearing supports. These results indicate that with an effective radial stiffness as shown below at each bearing set, the operating requirements can be met as follows:

K_{eff} = $0.06(10)^6$ lb/in. per bearing set

N_{1R} = 16,000 rpm)
)---shaft rigid body critical speeds
N_{2R} = 24,000 rpm)

N_{3B} = 92,000 rpm - shaft bending critical speed

N_{1R} and N_{2R} are rotor natural frequencies that are controlled mainly by effective stiffness (K) at the bearings and N_{3R} is controlled mainly by shaft bending stiffness (EI). These calculated critical frequencies meet the requirements of the design criteria which are:

N_{1R} and N_{2R} $<$ 30,000 rpm

N_{3B} $>$ 92,000 rpm

d Seals

The fuel TPA contains a static lift-off seal between the turbine and turbine end bearings, a "zero leakage-zero wear" relationship to achieve 300 starts and 10 hour life is required for the current application.

(b) Oxidizer Turbopump

1 Assembly

The conceptual design of the main oxidizer turbopump is shown in Figure 194. Although other types of low speed inducers can be used, this figure shows a low speed inducer upstream of the main pump which is driven by a "full flow" turbine located between the high speed inducer and first-stage impeller. The main pump is a "stage-and-a-half" design with the first-stage discharging 85.5% of the flow to the secondary injector and 14.5% to the half-stage which discharges its flow to the primary injector. The impellers are shrouded to minimize tolerance requirements for tight axial clearances and to provide bias loads for the single acting thrust balancer. The pump is driven by a single-stage partial-admission turbine.

The pump and turbine are mounted on a shaft which is supported by two sets of preloaded angular contact bearings. The turbine end bearings are cooled with liquid hydrogen and the pump and bearings are cooled with liquid oxygen. A single acting hydrostatic thrust balancer is located at the backside of the first-stage impeller. The balancer operates in conjunction with a bias load on the front shroud of the impeller and the operation of the balancer is similar to that described for the fuel turbopump. The pump end bearing outer races are spring-loaded axially toward pump suction so that they can take transient axial loads and prevent balancer rubbing, but still allow the balancer to operate during steady-state without overloading the bearings. The turbine end bearings are free to float axially and take radial loads only. The turbopump operating speed range is above the first and second criticals, but below the third (shaft bending). Both bearing sets are spring-loaded radially to provide a mechanism for "tuning" the spring rate and to provide damping during operation through the criticals. An interpropellant seal is located between the "half-stage" impeller and the turbine end bearings. The seal system consists of a static and dynamic seal for each propellant separated by a helium purge seal. The interpropellant oxidizer seal located on the backside of the second-stage impeller, consists of a hydrostatic face-type seal for dynamic operation with a return vent to the inducer inlet. A separate narrow inner-land serves as a static seal for engine pre-fire and post-fire and also as a close clearance sealing dam to the oxidizer overboard vent. The fuel interpropellant seal consists of a high pressure staged labyrinth adjacent to the turbine end bearing cavity. Between this high pressure seal and the low pressure stage is a lift-off sealing device with a radial mating face. This seal is held closed by bellows spring-force and actuated-open by the pressure forces from an external pressure source. A return vent to the first-stage

fuel impeller inlet downstream of the labyrinth allows a low pressure differential for the final seal stage before the overboard vent. This final stage is a fluid film floating journal seal downstream of the static sealing land. The helium purge seal, located between the oxidizer and fuel vents, is a fluid film floating journal seal. Helium is supplied to the central cavity of the seal and flows to the respective oxidizer and fuel cavities.

The low speed inducer and turbine shaft is located on its own bearings and is decoupled mechanically from the high speed shaft.

2 Oxidizer Pump

The impellers for the stages are arranged front to back with the high speed inducer and first-stage impeller outboard of the bearings. The first-stage has an axial inlet at the outboard end of the turbopump and both stages have shrouded impeller vanes. The first-stage vanes are backswept at the discharge and the half-stage has radial vanes. The first-stage has a diffuser vane type discharge housing with a single discharge pipe, and the second-stage has a vaneless diffuser volute type discharge.

Several alternate pump configurations were considered before this selection was made such as open impellers, back-to-back impeller orientations and reversed orientation impellers with a radial inlet. The pump configuration selected has a conventional impeller orientation that integrates well with the thrust balancer and bearings, and allows a housing configuration without high pressure external seals.

3 Oxidizer Turbine

A single stage partial admission design was selected for the oxidizer TPA since it is the simplest design mechanically and delivers a satisfactory efficiency (52% at U/C = 0.25). Although higher efficiencies can be achieved with multiple staging, the possible gain in cycle efficiency was considered negligible. The turbine design mean blade speed was selected to be 1100 ft/sec to achieve oxidizer turbopump startup times comparable to the fuel turbopump.

4 Power Transmission

a Rotor Support

Similar to the fuel TPA, two important operational characteristics which exist for the rotor and bearing system are axial thrust and rotor system dynamics. Axial thrust in the oxidizer TPA has been balanced by a system of pressurized areas, rolling element bearings (for transient rotor axial positioning), and a self-compensating hydrostatic thrust bearing on the backside of the first stage impeller. Typical performance characteristics for a hydrostatic thrust

balancer in its nominal operating position is to provide a force towards the pump inlet. Also, the net thrust from the turbine is towards the pump inlet. These thrust loads are resisted by the net thrust on the second-stage impeller and the thrust on the front shroud of the first-stage impeller. Since the turbine bearing cavity contains fuel at high pressure and this pressure acts over a relatively large area in the vicinity of the turbine lift-off seal, the axial thrust balance is dependent on the fuel pump pressures. The pressure in this cavity must be approximately 2000 psi to achieve a thrust balance. Several configurations of the oxidizer TPA have been investigated for critical speed. Two approaches can be taken for the oxidizer TPA rotating system design: (1) to operate below the first system critical speed using a stiff bearing system, and (2) to operate below the third system critical speed (shaft bending critical) on soft mounted bearings. Due to the high operating speed, large turbine diameter relative to the pump impeller diameters, bearing stiffness characteristics, and the throttling range requirements, operating below the shaft bending critical speed on "soft" bearings or "soft mounted" bearings was considered to be the better choice of rotor support.

<u>b</u> Oxidizer TPA Bearings

A preliminary analysis and selection of bearing sizes and configuration was made for the oxidizer turbopump design. 106 size "DB" ground duplex ball bearings were selected for the turbine end, and 104 size "DB" ground duplex ball bearings were selected for the pump end. The rotating speed (50,000 rpm) is acceptable for these sizes.

Bearing Location	Bore Size	DN
Pump	20	1.0×10^6
Turbine	30	1.5×10^6

Aerojet has successfully operated bearings in LO_2 at DN values up to 1.9×10^6. The 0.2 factored B_1 fatigue lives of the bearing preliminary designs appear adequate. The pump end bearings of the turbopump are most sensitive to axial thrust load. This load shall not exceed 78 lb (25 lb per load plus 53 lb applied load) to achieve a 0.2 factored B_1 life of 10 hours. The radial loads shall be limited to 10 lb from the turbine and 100 lb from the pump. A life reduction factor of 0.2 is applied to the B_1 lives to account for the low viscosity coolant. The predicted 0.2 B_1 bearing lives for each of the four bearings starting from the pump end are 10.8, 163.8, 150.2 and 146.3 hours. More analysis is necessary to further optimize bearing internal geometry to increase life.

c Critical Speed

A preliminary critical speed analysis for the oxidizer turbopump (Figure 194) was conducted using the same computer program as described for the fuel turbopump used.

The results of the rotor-only parametric analysis are shown in Appendix A, where the first three rotor critical speeds are plotted versus effective radial stiffness at the bearing supports. These results indicate that with an effective radial stiffness as shown below, at each bearing set, the operating requirements can be met as follows:

$$K_{EFF} = .03(10)^6 \text{ lb/in. per bearing set}$$

$$N_{1R} = 12{,}000 \text{ rpm}$$

$$N_{2R} = 16{,}000 \text{ rpm}$$

$$N_{3B} = 64{,}000 \text{ rpm}$$

N_{1R} and N_{2R} are rotor natural frequencies that are controlled mainly by effective stiffness (K) at the bearings and N_{3B} is controlled mainly by shaft bending stiffness (EI). These calculated critical frequencies meet the requirements of the design criteria which are:

$$N_{1R} \text{ and } N_{2R} < 17{,}000$$

$$N_{3B} > 57{,}500$$

d Seals

The oxidizer TPA requires an interpropellant seal to separate the LO_2 in the pump end from the hot GH_2 rich gas at the turbine end. LH_2 was selected to lubricate the turbine end bearings rather than LO_2, which prevents GO_2 from entering the turbine exhaust. The interpropellant seal of the oxidizer TPA (Figure 199) is required to separate LO_2 and LH_2. A static lift-off seal is located between the turbine and bearing and the turbine. This seal is similar in design and operates identical to the lift-off seal for the fuel turbopump.

Staged hydrostatic seals are used in the interpropellant seal to control LO_2 and LH_2 overboard leakage. This design vents the high pressure from the high pressure "half-stage" impeller and the turbine end bearing cavities to the inlets of the respective main pumps, and then allows a controlled amount of leakage from the reduced pressure to an overboard vent. This overboard leakage could be used for propellant tank make up pressurant flow. A helium purge is provided between the oxidizer and fuel overboard vents and the amount of helium is minimized by

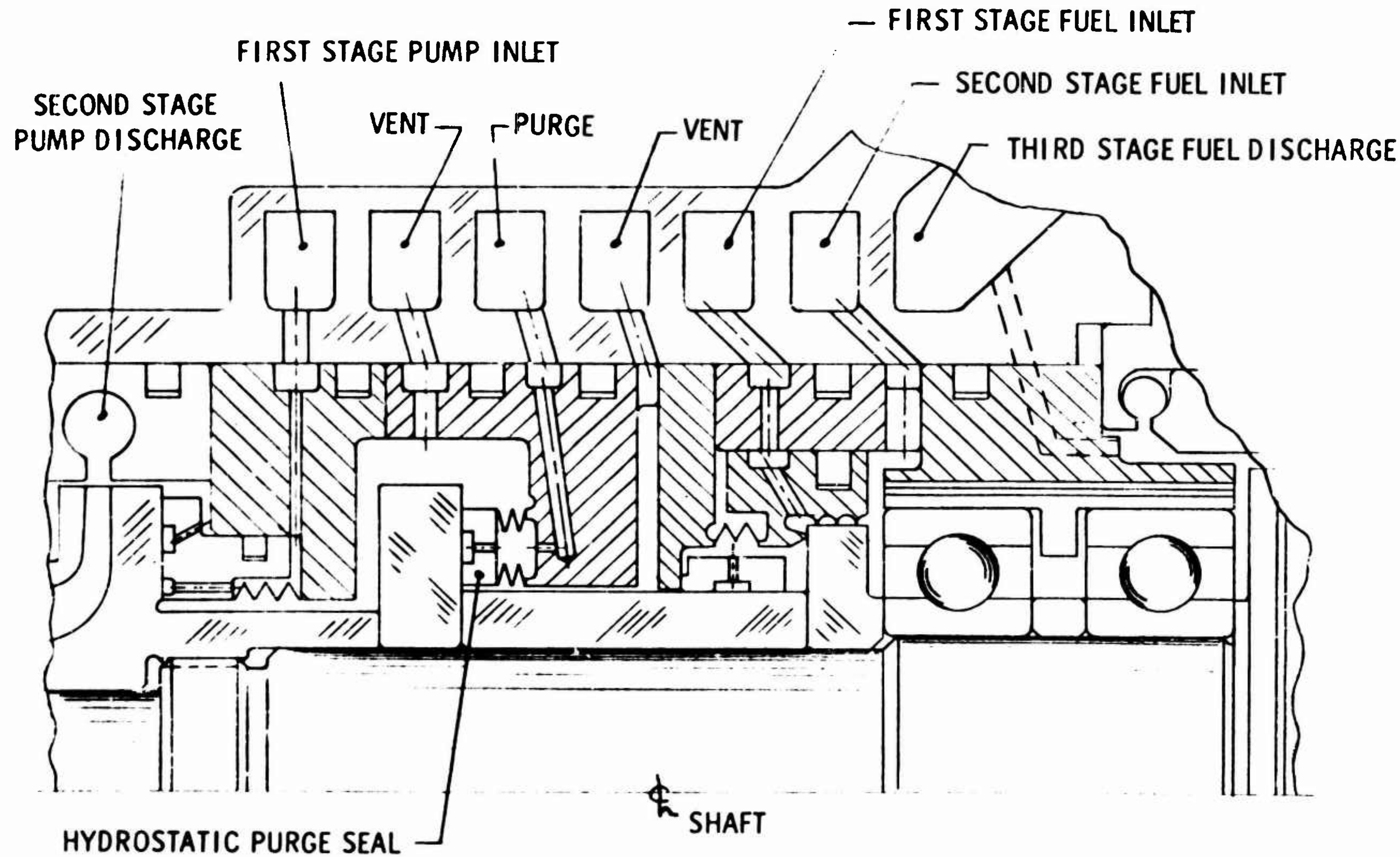


Figure 199. Oxidizer Turbopump Interpropellant Seal

the use of hydrostatic helium gas seal. An alternate interpropellant seal concept could be adapted to the oxidizer turbopump as configured. This concept takes advantage of the vacuum environment at which the turbopump will normally operate. The helium purge seal provision, as shown in Figure 199, would be eliminated and a low restriction overboard vent cavity between the GO_2 and GH_2 vent cavities would be provided. The low restriction overboard vent cavity would be exposed to the vacuum of space and achieve a pressure value sufficiently low in the seal cavity to assure non-ignition of a 0.005 lb/sec GH_2/GO_2 vapor mix.

(c) Low Speed Pumps

In order to achieve a minimum weight, high efficiency turbopump, low speed boost pumps are required to permit the designing of the main turbopumps to operate at high speed. The low speed boost pumps raise the pressure level of the propellant from the system supplied minimum NPSH values to the pressure level required by the main pump high speed inducer.

The design configuration selected for the baseline TPA design consists of a low speed inducer which is driven by a full flow hydraulic turbine located downstream of the high speed inducer, Figure 188 and Figure 194, for the fuel and oxidizer TPA's, respectively. The inducer and turbine are attached to a common shaft which is supported by propellant cooled preloaded ball bearings. The utilization of a low speed boost pump permits the optimization and selection of the shaft speed of the main turbopump independent of engine system allocated NPSH values. The boost pump shaft speed and design can in turn be adjusted to accommodate the specified NPSH values.

The baseline low speed boost pump design and operating parameters are given in Table LI for the baseline NPSH value of 16 and 60 feet for the oxidizer and fuel suction, respectively. The impact of designing the boost pumps to operate down to NPSH values equaling zero are relatively small (See section below - Design Selection - Low Speed Boost Pumps) which suggests that the propellant speed system design requirements be zero feet NPSH suction capability at the pump suctions.

(d) Performance

1 Design Point

Design point, performance, and design dimensions for the low and high speed turbomachines are shown in Table LI for the fuel and oxidizer turbopump, Figures 188 and 194, respectively.

2 Off-Design

a Pumps

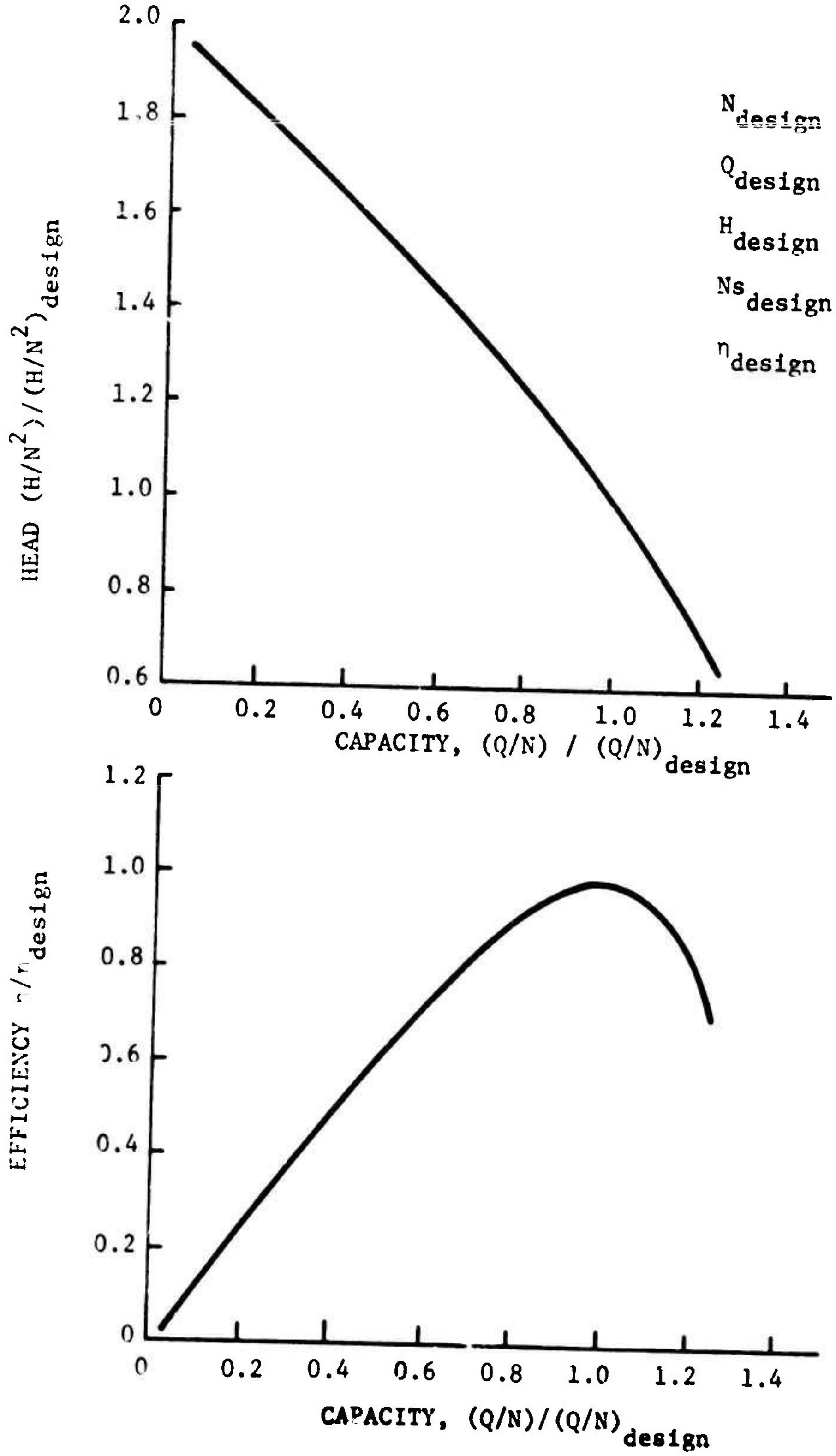
Engine system studies show for 5:1 throttling and operation between mixture ratio limits that the pumps will be required to operate from 105% to 50% of design flow coefficient (expressed as $(Q/N)/(Q/N)_{design}$) in the main pumps, 102% and 25% of design flow coefficient in the oxidizer half-stage pump, and 102 to 46% of design flow coefficient in the low speed inducer. The lowest flow coefficient operation occurs at the engine 5:1 throttled operation. The pump normalized head - capacity characteristics for the low speed inducers, high speed inducers and high speed main stages are given in Figures 200 through 202 for the fuel and oxidizer turbopumps. Correspondingly, the pump shaft speed was reduced to 37% of rated in oxidizer TPA and 59% in the fuel TPA at the 5:1 throttle point. This reduced shaft speed, at the off-design reduced flow coefficient operation, reduces the absolute values of axial and radial forces that may be generated by the pump; the pump axial unbalance forces being compensated by the axial thrust balancer. The primary concern in operating the pumps at reduced flow coefficient is the possible operation of the pumps near their stall point. The pump designs selected accommodate the required off-design flow variations. The design areas of accommodation are the impeller blade exit angles and the diffuser design.

The suction performance of the low and high speed inducers and the main stage impellers is shown in Figures 191 and 195 for the fuel and oxidizer pump, respectively. The off-design operation is shown in terms of design flow, the ratio of operating Q/N to the design value of Q/N for engine operation at thrust values of 25 to 5K. The operating suction specific speed is shown as the ratio of design suction specific speed. The percent head loss for each pump element can be determined from the location of the respective operating points and the lines of constant percentage head loss. It can be seen that the low and high speed inducers will experience no head loss over the specified engine throttle range. The first-stage of the fuel pump main stage will experience approximately a two percent and the oxidizer pump a one percent head loss at the extreme engine throttle point (5K).

b Hot Gas Turbines

The high speed main turbine off-design performance can be approximated by the equation:

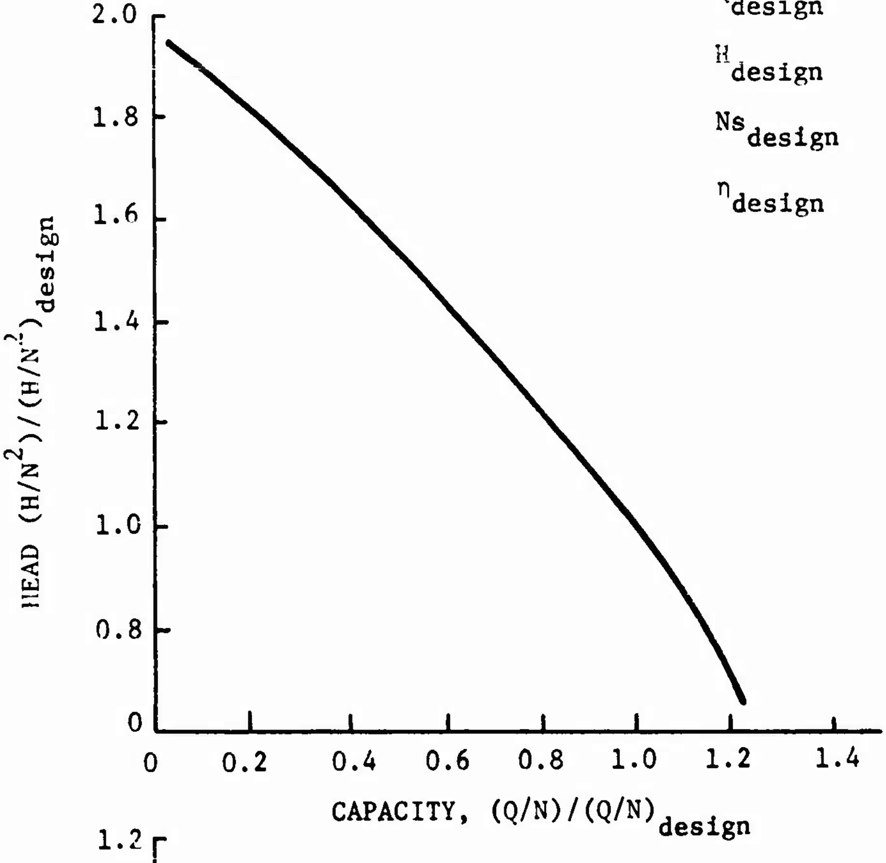
$$\frac{\quad}{\quad_{design}} = \left[2\left(\frac{\left(\frac{U_m}{c}\right)}{\left(\frac{U_m}{c}\right)_{design}} \right) - \left(\frac{\left(\frac{U_m}{c}\right)}{\left(\frac{U_m}{c}\right)_{design}} \right)^2 \right]$$



	OXIDIZER	FUEL
N_{design}	14286	22857
Q_{design}	290	792
H_{design}	239	874
Ns_{design}	4000	4000
η_{design}	66	71

Figure 200. Normalized Pump Parameters, Boost Pumps

	OXIDIZER	FUEL
N_{design}	50,000	80,000
Q_{design}	290	792
H_{design}	1,272	4,768
Ns_{design}	4,000	3,923
η_{design}	66	71



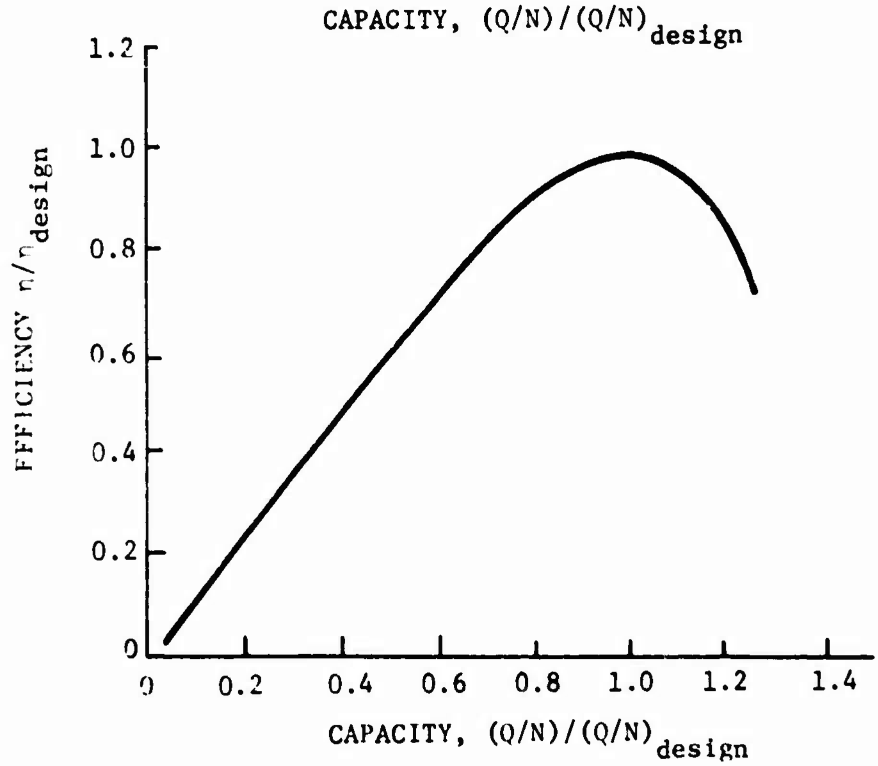
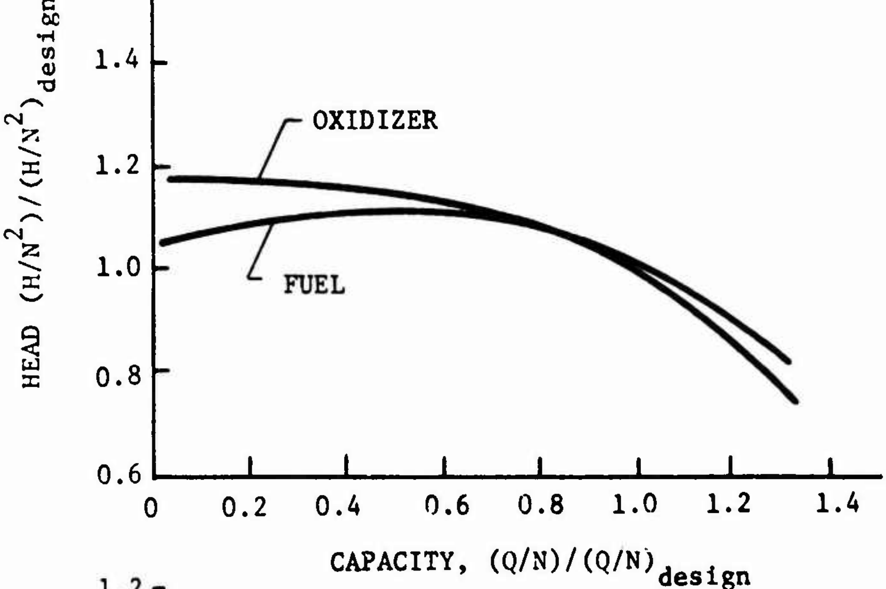


Figure 201. Normalized Pump Parameters, High Speed Inducers

	OXIDIZER	FUEL
N_{design}	50,000	80,000
Q_{design}	290	792
H_{design}	5,173	137,206 (Total in Three Stages)
Ns_{design}	1,397	720
η_{design}	64	61.5



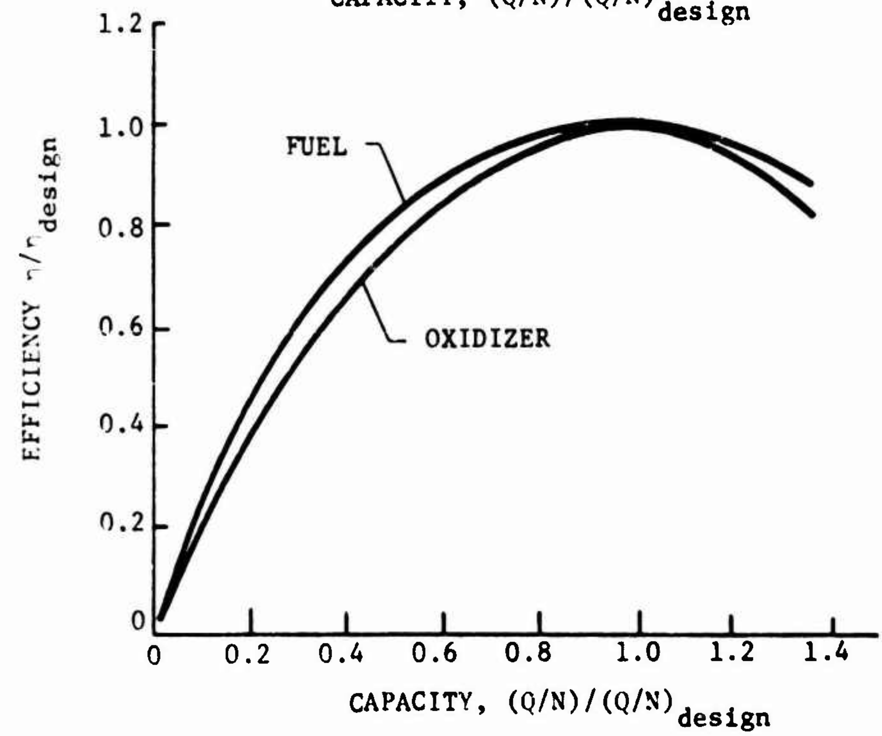


Figure 202. Normalized Pump Parameters, Pumps Main

c Hydraulic Turbines

Estimating the performance of the full flow hydraulic turbine necessitates defining the fluid velocity leaving the high speed inducer that drives the low speed hydraulic turbine, in that there is no nozzle segment directing the flow to the turbine. The turbine off-design efficiency can be approximated by the same equation used for the hot gas turbine above. The fluid velocity leaving the high speed inducer is approximated from the design head coefficient ψ, flow coefficient ϕ, impeller mean blade speed U_m and efficiency η values of the high speed inducer given in Table . The absolute velocity is given by

$$C_{ab} = \left(C_m^2 + C_u^2\right)^{1/2} \text{ ft/sec}$$

where meridonial velocity

$$C_m = \left(U_m \times \phi\right) \text{ ft/sec}$$

tangential velocity

$$C_u = \left(\frac{U_m \times \psi}{\eta_{\text{Ind Diffuser}}}\right) \text{ ft/sec}$$

The off-design flow coefficient ϕ can be computed from the design point flow coefficient ϕdes

$$\phi = \phi_{des} \frac{(Q/N)}{(Q/N)_{des}}$$

The off-design head coefficient ψ can be computed from the design point head coefficient ψdes

$$\psi = \psi_{des} \frac{(H/N^2)}{(H/N^2)_{des}}$$

It should be noted that this method of approximating turbine off-design efficiency neglects the change in fluid angle as it approaches the turbine; however, predicted speed ratio at off-design condition of the integrated low speed boost pump and high speed main pump agrees within 8% of test data.

A further simplification of low speed boost pump modeling would be to assume that the ratio of shaft speed main to boost remain constant over the operating range. This method would give a boost pump shaft speed that is 15% low for the 5K throttled thrust level.

(e) Weight

A preliminary weight analysis was performed of the baseline fuel and oxidizer turbopumps shown in Figures 188 and 194, respectively. The calculated weight values are given in Table LIII and are based on materials of construction discussed in the section below headed Materials.

(f) Materials

1 Turbine Housings

The oxidizer and fuel turbine housings are welded assemblies of ARMCO 22-13-5 stainless steel. This alloy is selected on the basis of its resistance to hydrogen embrittlement, high strength and fabricability. ARMCO 22-13-5 possesses the highest strength of the nonheat-treatable stainless steels, which are not susceptible to hydrogen embrittlement. The use of the higher strength, heat-treatable alloy, A-286 stainless steel, is avoided due to its poor weldability. ARMCO 22-13-5 also possesses equivalent elevated temperature strength and near-equivalent ductility to the nonhardenable nickel base alloys such as Hastelloy X or Inconel 625 for temperatures to 1400°F. The nickel base alloys would normally be used in the application except for their susceptibility to hydrogen embrittlement.

ARMCO 22-13-5 has excellent weldability except that nitrogen outgassing of the weld was experienced during initial electron beam welding at ALRC. However, multi-pass weld procedures were established to alleviate this problem.

2 Turbine Rotor Blades and Stator Vanes

The turbine rotor service temperature and stress dictate the use of heat-treatable nickel base alloys which are susceptible to hydrogen embrittlement. The preburner exhaust is an embrittling environment as evidenced by recent testing conducted at ALRC which indicated that the water vapor content of the hydrogen-rich gas does not inhibit embrittlement. The turbine blade steady-state temperatures are in a range where embrittlement effects are minimal. However, the turbines are required to undergo transients through the embrittlement temperature range, and this has been taken into consideration in the design analysis. Consideration has also been given to the more severe embrittling environment in the turbine disc area where temperatures are lower.

Waspalloy is selected as the stator vane and turbine material for both the oxidizer and fuel turbopumps. This alloy is selected over the higher strength cast alloys since its higher ductility will provide improved low cycle fatigue-life in areas of high thermal stress. The turbine rotors will be machined from forgings to provide an integral blade-disc configuration.

TABLE LIII

25K OOS TURBOPUMP WEIGHT BREAKDOWN

	Oxidizer	Fuel
Boost Turbopump - Pounds		
Impeller	0.40	0.47
Shaft and Bearing Inner Race	0.63	0.68
Hydraulic Turbine	0.06	0.10
Housing and Bearing Outer Race	3.44	4.02
TOTAL	4.53	5.27
Main Turbopump - Pounds		
Inducer	0.22	0.31
Impeller(s) Main	0.25	3.46
Impeller (Half Stage)	0.28	-
Shaft and Rotating Elements	5.77	1.83
First Stage Turbine Rotor	1.98	0.94
Second Stage Turbine Rotor	-	0.94
Pump Housing and Power Transmission Housing	12.44	21.86
Turbine Nozzle Assembly	4.69	2.72
TOTAL	25.63	32.06
Boost + Main Turbopumps - Pounds		
TOTAL	30.16	37.33

3 Shaft

The oxidizer and fuel shafts will be fabricated from A-286 material which is the highest strength embrittlement-immune material available.

4 Pumps and Power Transmission Housing

The main fuel pump housing, impellers and power transmission housing will be fabricated from 5Al-2.5Sn-ELI Titanium alloy. This material is selected on the basis of its strength, excellent fabricability and low temperature ductility. Titanium and its alloys are subject to hydrogen embrittlement, however, recent tests at ALRC indicate that embrittlement is not significant at temperatures below approximately 0°F. Since the pump will be chilled at engine start, hydrogen embrittlement is not a concern for these parts.

Candidate materials for the main oxidizer pump housing include ARMCO-22-13-5 and Inconel 718 with ARMCO-22-13-5 being selected since the turbine end of the housing is exposed to a hydrogen embrittlement environment (Turbine Exhaust Gas) and ARMCO-22-13-5 is not subject to hydrogen embrittlement. Titanium cannot be used in this application as in the hydrogen main pump due to its poor compatibility with liquid oxygen. Candidate materials for the high speed impeller include Inconel 718 and Aluminum 7075-T73 with Inconel 718 being selected for its greater strength.

The fuel and oxidizer boost pump impellers and hydraulic turbines will be made from Aluminum 7075-T73. The housings will be made from Aluminum 6061-T6 to take advantage of its weldability properties.

5 Bearings

Stainless Steel 440C is selected as the bearing material. This material is subject to hydrogen embrittlement, low temperature embrittlement and stress corrosion cracking. This will not adversely affect the performance of this alloy due to the cryogenic service temperature and the compressive nature of loading. The stress corrosion susceptibility is a concern due to possible high assembly stresses since it possesses a lower coefficient of expansion than the shaft material (A-286) and will require a shrink-fit to provide proper clearances at the lower operating temperature.

h. Turbine Hot Gas Manifold

(1) Description

The baseline hot gas manifold design is defined by subassembly Drawing 1161575 presented as Figure 203. Within the basic envelope, a primary goal in the detail design study was to achieve a simple and reliable uncooled hot gas manifold design. Regeneratively-cooled or transpiration-cooled concepts were not found necessary at a turbine temperature of 1860°R.

The primary function of the hot gas manifold is to contain fuel rich hot gas (FRHG), emanating from the single preburner at 1860°R, in a common plenum chamber, duct the FRHG through the individual FTPA and OTPA turbines in parallel, and finally direct the FRHG to the main injector to complete staged combustion with primary oxidizer flow. It is also the primary structural component on the engine supporting the TCA and powerhead. Two distinct areas of design criticality exist in the hot gas manifold design; (1) the turbine and preburner arrangement for turbopump drive power requirements, and (2) the collection of turbine discharge gases and manifolding to the main injector for staged combustion with the primary oxidizer flow. For this reason, the technical discussion of the hot gas manifold has been separated appropriately into two distinct sections; turbine hot gas manifold and injector hot gas inlet manifold. Technical discussion of the injector hot gas inlet manifold can be found in detail in Section III,B,2,a. A brief discussion is presented herein to keep the context of the total hot gas manifold design.

(a) Injector Hot Gas Inlet Manifold

The hot gas inlet manifold has several functions. Physically, it provides a mounting pad for the engine gimbal block assembly, mates with the main injector, and provides structural support for the turbine hot gas manifold, FTPA, OTPA and preburner combination. It receives FRHG from the fuel and oxidizer turbine exhaust ducts during operation, and from a fuel coolant jacket and turbine bypass line during starting. It distributes these gases properly to the injector inlet in a minimum length envelope, with minimum pressure and thermal losses. Provision is also made in the FRHG inlet manifold design for a thrust chamber igniter. The manifold life must be equal to or greater than that of the main injector in order to optimize engine maintenance procedures.

(b) Turbine Hot Gas Manifold

The turbine hot gas manifold has several functions. Physically it provides mounting flanges for the fuel turbopump assembly (FTPA), oxidizer turbopump assembly (OTPA) and single preburner assembly. The primary function, as previously stated, is to contain the fuel rich hot gas (FRHG), emanating from the mounted single preburner at 1860°R, in a common plenum chamber and duct the FRHG through the individual FTPA and

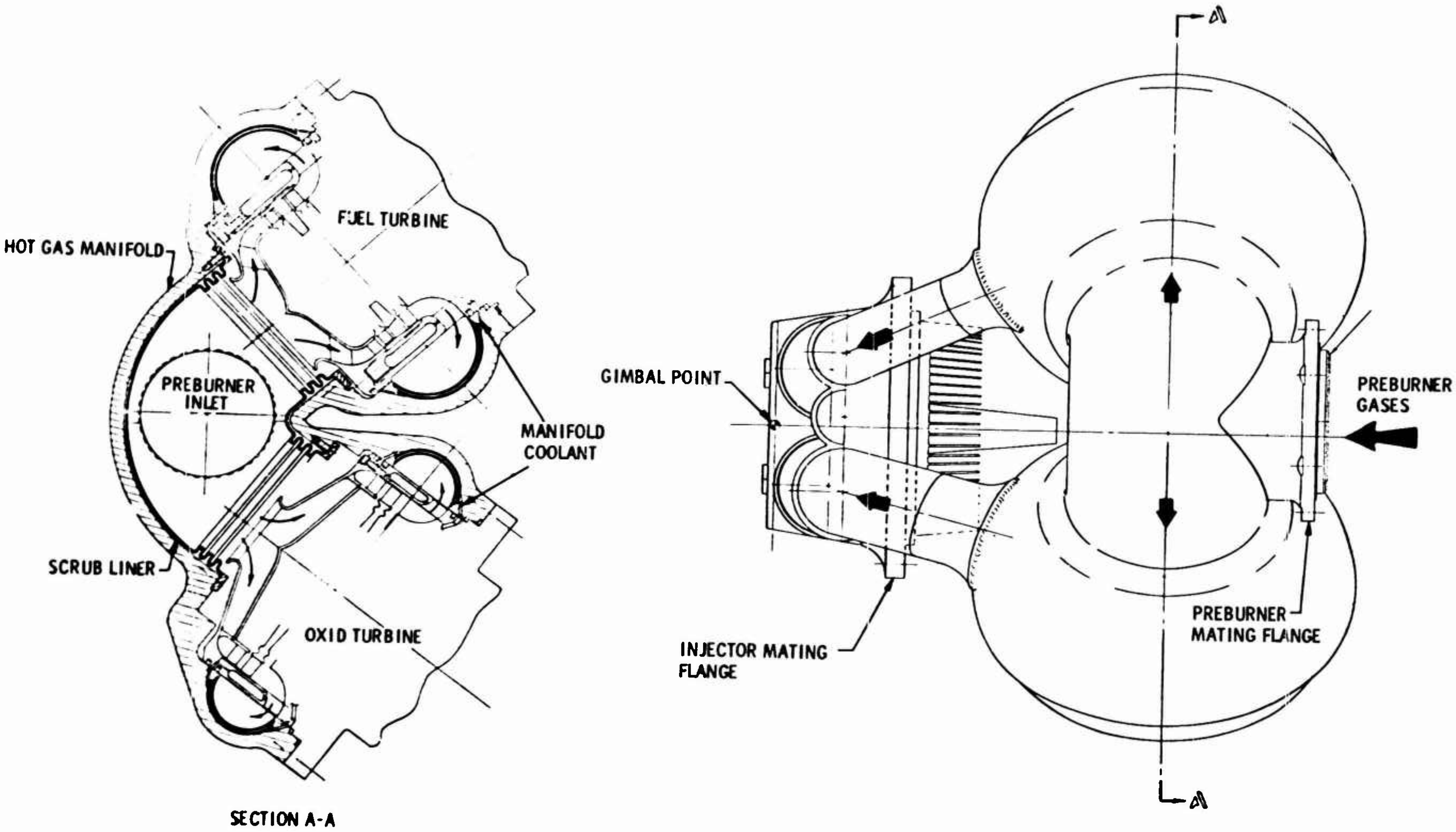


Figure 203. Hot Gas Manifold

OTPA turbines in parallel providing turbopump drive power. Turbopump mounting and turbine gas ducting are shown on the engine assembly drawing.

The turbine hot gas manifold is composed of three (3) basic units which are integrated into a common structure. The units are: (1) plenum chamber, (2) fuel turbopump turbine housing, and (3) oxidizer turbopump turbine housing.

ARMCO 22-13-5 was selected exclusively for use in the hot gas manifold design because of turbine temperature excursion from 1860°R nominal down into the hydrogen embrittlement range of 210 to 1700°R during engine throttling. In keeping with the basic ground rules to achieve an uncooled manifold design evaluation, ARMCO 22-13-5 indicated a satisfactory stress margin providing the wall thickness was commensurate with operating 1860°R turbine temperature and turbine inlet and exit pressures of 2962 psia and 1995 psia.

(2) Low Cycle Fatigue (LCF) Life Design Impact

The engine requirement which has considerable impact on the hot gas manifold design is thermal cycle life requirements. The hardware failure mode encountered with long life combustion components is metal fatigue cracking. This type of failure occurs when the metal is repeatedly strain cycled beyond its elastic limit, such that permanent deformation occurs without single load cycle material fracture. A detailed discussion of thermal stress conditions and corrective measures, which can be utilized by the designer, is presented in Section III,B,2,a, Combustion Components.

Of primary importance was to ensure the hot gas manifold design would meet the total engine thermal cycle/life requirements imposed by the contract summarized as follows:

	Thermal Cycles	Life (Hours)
	300 Initial	10
	300 1st Overhaul	10
	300 2nd Overhaul	10
	300 3rd Overhaul	10
	300 4th Overhaul	10
Total	1500	50

Capability to operate over the total thermal cycle/life span of 1500 cycles/50 hours was considered mandatory from an engine maintenance standpoint, since removal of the hot gas manifold essentially required a complete engine disassembly.

To meet the thermal cycle/life criteria above, required the addition of the following thermal barrier devices:

Scrub liners, fabricated of ARMCO 22-13-5 sheet, were placed in the plenum chamber and preburner transition section as shown in Figure 204.

Cooling passages in fuel and oxidizer turbopump turbine housings for utilizing hydrogen leakage to cool both housings.

Uncooled scrub liners are utilized in the prescribed areas to reduce the ΔT between the combustion gas and heavy structural outer wall during the start transient, thus maintaining thermal shock at a minimum. The scrub liners, in conjunction with the stagnant gas between the liners and structural wall, serve to reduce the heat flux by eliminating forced convection to the structural wall. The ARMCO scrub liners are uncooled since the turbine temperature of 1860°R is considerably lower than the melting temperature. The scrub liners are freed of mechanical loads induced by thermal growth by bellows sections in the plenum chamber and are unrestrained in the preburner transition section.

Fuel and oxidizer turbopump housings also utilize ARMCO 22-13-5 scrub liners in conjunction with a gas annulus. Additionally, available hydrogen leakage of approximately 0.25 lb/sec from the turbopump seals is directed through coolant passages and discharged into the turbine exhaust gases in each turbine housing as shown in Figure 204. Cooling the turbine housings with hydrogen leakage serves to reduce the thermal gradient between the cold turbopump mounting flanges and hot gas manifold turbine housings to an acceptable level.

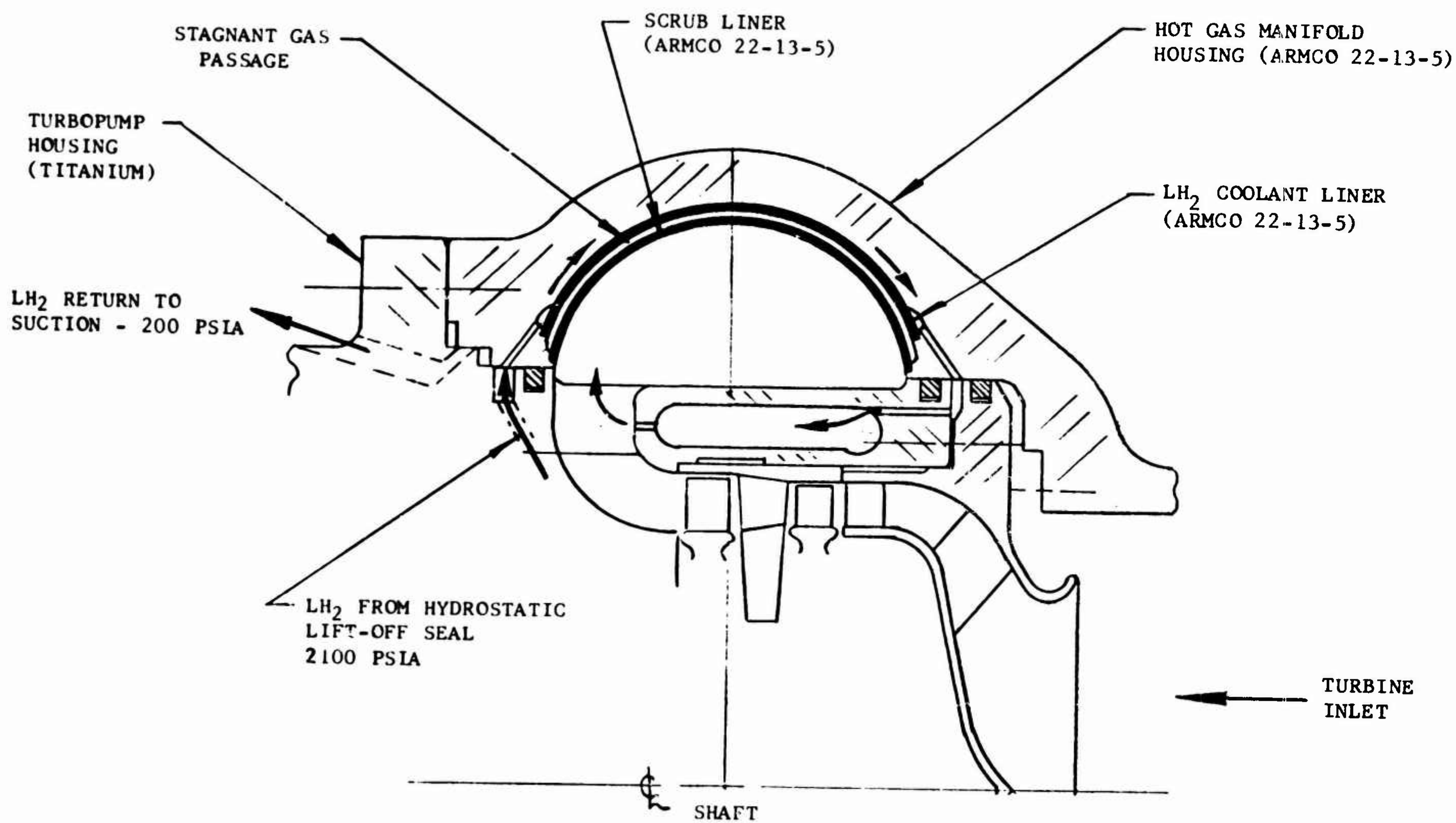


Figure 204. Hot Gas Manifold Loading Concept

i. Propellant Lines

(1) Suction Lines

Suction line definition was established as vaned elbows attached to the laterally-mounted boost pump inlets at an engine vehicle station plane 4.50-in. aft of the gimbal point. As discussed in Section III,B,1,d,(2),(b),1, this interface definition provides flexibility to vehicle contractors for suction line routings utilizing either pressure volume compensating bellows (PVC's) or gimbaled suction lines attached to the vaned elbow inlets. It is anticipated that final selection of suction line routing will be made by the customer and the vehicle contractor. ALRC will then proceed with final design and procurement of either PVC's or gimbaled suction lines. Since both designs are considered candidates both are discussed herein.

(a) Pressure Volume Compensating Bellows (PVC's) vs Gimbaled Suction Lines

As shown in the suction line matrix, Section III,B,1,d,2, minimum engine dry and wet weights can be achieved with PVC's. The ΔP effect on vehicle NPSH is lower on the oxidizer circuit and comparable on the fuel circuit when comparing PVC's to gimbaled suction lines. The cost differential of $16,000 a set for PVC's, as compared to $9,000 a set for gimbaled suction lines, must be considered, but may be offset by the improved performance and weight data.

Arrowhead Products Division of Federal Mogul Corporation, Los Alamitos, California, was contacted and preliminary design requirements established on PVC, gimbaled suction line, and thermal expansion joint design. The cost and weight data established in Section III,B,1,d,2 was accomplished through coordination with Arrowhead. Arrowhead's experience in gimbaled lines manufactured for ALRC on Apollo, Transtage and Titan contracts, as well as their experience on PVC design for the F-1 engine and other applications, made them the logical source for propellant line design coordination.

(2) Fluid Interconnect Line and Flange Design

(a) Semirigid Lines

Semirigid lines shown connecting the major assemblies on the engine assembly drawing in Section III,B,1,d,1 do not have bellows. The required flexibility for expansion, contraction, and differential movement is obtained by line routing. The major lines of this type on the engine are.

1. Fuel discharge line.
2. Fuel bypass line.
3. Preburner fuel supply line.
4. Igniter fuel supply line (TCA igniter only)
5. Oxidizer discharge line.
6. Preburner oxidizer supply line
7. Igniter oxidizer supply line (TCA igniter only)

It is anticipated that expansion joints will be required at strategic areas in some lines to compensate for thermal growth, particularly in relation to the hot gas manifold. Exact location and design definition must be accomplished by detailed structural modeling of the engine.

Most of the engine interconnecting lines are of the semirigid type. Material is Inconel 718 for all lines except those subjected to heated hydrogen. These are the fuel preburner supply line connecting the chamber coolant jacket outlet and preburner assembly, and the autogenous fuel pressurization lines to the vehicle fuel tank.

(b) Flexible Fluid Interface Lines

Although not shown on the engine assembly drawing, the small propellant recirculating lines and the helium purge lines engine/vehicle interface is established at a flat station plane 4.50 in. aft of the gimbal point. Physical plumbing of these lines to this station would make them of the semirigid type since relative motion with respect to the engine gimbal point would require flexible features in the vehicle lines. It is anticipated, however, that final selection of the suction line interfaces by the customer and the vehicle contractor will move the actual engine interface beyond the existing envelope, set at the plane through the gimbal point, to the inlets of the selected PVC's or gimbaled suction lines. Relocation of the suction line engine/vehicle interface would logically move the small propellant recirculating lines and helium purge lines to the same station plane. In this case, the lines will become flexible since relative engine motion across the gimbal plane will require flexible features. Braided bellows will then be used in all flexible fluid interface lines which connect gimbaling parts of the engine to the vehicle to provide the required flexibility. These lines carry less than 1500 psia and are well within the state-of-the-art.

(c) Line Insulation

All fuel lines above the main fuel shutoff valve and to the coolant jacket inlet are insulated to mimimize propellant consumption during engine chilldown and idle mode operation. This insulation is comprised of a stainless steel jacket around the lines. The annular volume between the line and the outer shell is purged and filled with CO_2 at a positive pressure of 0.5 psig. During chilldown, the CO_2 condenses producing a cryopumped vacuum insulation to inhibit formation of liquid air and to prevent excessive boil-off. Owens Corning AA-GHF Fiberglass batting is installed in the annulus to provide insulation redundancy and to minimize the heat loss through CO_2 crystals which could accumulate in the lower regions of the lines. The batting reduces the gas volume in the annulus which, in turn, enhances the rate of cryo-pumping. A port is located in one end of the duct for checkout and maintenance of the CO_2 gas in the insulation jacket. A burst-disc is incorporated in the outer shell of the line so that the line will not rupture in the event of hydrogen leakage into the annulus.

(d) Separable Fluid Connections

Separable fluid connections are incorporated in the OOS Engine at strategic locations to permit the easy replacement of components. It is important to minimize the number of separable connections because each is a source of leakage and adds weight to the engine system. The separable fluid connections were designed to satisfy engine system requirements with the lightest possible weight and include the following key features:

1. Cantilevered flanges which permit minimum weight and provide the best environment for the seal (minimum deflection at the sealing interfaces).

2. Flat interface requiring minimum separation of the flanges to remove components and lines as well as to properly install the seal.

3. Positive contact between flanges to guarantee electrical continuity and assure that no electrical potential is possible between the flanges.

4. All-metal seal design capable of being used at all temperatures expected in the engine system thus eliminating non-interchangeability problems.

5. Spherical bearing surfaces at the flange/fastener interfaces to prevent bolt bending caused by normal cantilevered flange rotation as well as misalignment between mating flanges.

6. Minimum bolt diameters which reduce the bolt circle diameters and thereby reduce weight.

7. Through-bolts, where possible, to reduce problems associated with flange misalignment and to enhance space maintenance capability.

8. Self-locking (all metal) nuts for easier and faster replacement of components by eliminating less desirable, more time-consuming locking methods (i.e., safety wiring).

1 Design Description

The separable fluid connections consist of simple bolted cantilevered flanges with the pivot located at the connection ID and the seal located between the pivot and the bolt circle. The fastener system consists of the following:

1. Standard 12-point high strength bolts.

2. All-metal 12-point self-locking high-strength nuts.

3. Spherical washers under the bolts and nuts with matching spherical surfaces in the flanges.

4. Preload Indicating (PLI) washers for improved preloading accuracy where significant weight savings will result from their use.

This design provides the best environment for the seal because the ID pivot is preloaded to a value equal to or greater than the separating loads produced by internal pressure. Therefore, the major deflection at the sealing edge during operational loading is that associated with relieving the load on the pivot.

The pivot bearing area is designed to prevent permanent deformation during preloading and thus maintains the proper squeeze on the seal to obtain seal preloading. A minimum of four slots (0.60-in. wide by 0.020 to 0.030-in. deep) are machined radially across the pivot face to assure the seal cavity is pressurized during leak checks and proper seal installation is verified.

The Harrison Manufacturing Company K-seal was selected for use in the separable fluid connections. The design is a modification of an ALRC standard seal that provides adequate sealing loads and has proven capable of satisfying the leakage requirements in a 3.80-in. connection tested as part of the ALRC IR&D Program, "High Pressure Fluid Connections". The NAFLEX seal was selected as the backup design to ensure that flange seal development problems do not pace the engine program. Flanges using the Harrison K-seal can be reworked to accept the NAFLEX seal. The seal configuration and test success experienced on the XLR-129 will also be considered.

j. Gimbal System

(1) Introduction

The gimbal system consists of the gimbal assembly (block) and the actuator clevises, as shown in Figure 205, are mounted in the gimbal planes to minimize actuator stroke and "cross-talk" between pitch and yaw axes.

The gimbal assembly for the OOS Engine provides a pivot point for gimbaling the engine with respect to the vehicle. The gimbal assembly transmits thrust load and side loads from the engine to the vehicle during engine operation and sustains the weight of the engine in static or nonoperative conditions.

The gimbal design selected for the OOS Engine utilizes two cylindrical surfaces 90° to each other with different radii and a common radius gimbal center external to the unit as shown in Figure 205 to obtain minimum engine length--a primary goal for the selected fixed nozzle engine design. The selected design which draws heavily upon experience gained in the Titan family of engines--where a similar gimbal is used--meets interface requirements, is more compatible with the conical center housing of the hot gas manifold, and has low development risk.

Gimbal actuators, although not assumed part of the engine system, were analyzed for engine/vehicle system definition and discussed from a system design standpoint along with the gimbal assembly in the following discussion.

(2) Design Description

(a) Gimbal Assembly

The gimbal assembly is shown in Figure 205. The gimbal assembly is comprised of seven major pieces, the upper gimbal block, pivot, lower gimbal block, two upper retainers and two lower retainers. The gimbal assembly moves on two pairs of cylindrical surfaces whose centerlines are at 90° to each other with a common radius center (gimbal center) but different radii. The upper surface of the pivot and the upper surface of the lower gimbal block are covered by a 0.012-in. layer of filled polytetrafluoroethylene (PTFE) material bonded to 7075-T73 aluminum material. All surfaces except those covered with PTFE are anodized. The surfaces on the upper gimbal block and the pivot which mate with the PTFE are lapped to a 6 to 8 rms finish. In addition to the main surfaces, the shear surfaces of the pivot and lower gimbal block have PTFE surfaces with matching retainer surfaces lapped to 6 to 8 rms. For side loads, the lower gimbal block and pivot have aluminum strips with PTFE surfaces bolted to the vertical sides. The upper gimbal block and pivot have lapped surfaces to match these surfaces.

The nonmetallic surface of filled PTFE sliding on a high finish hardened metallic surface has been

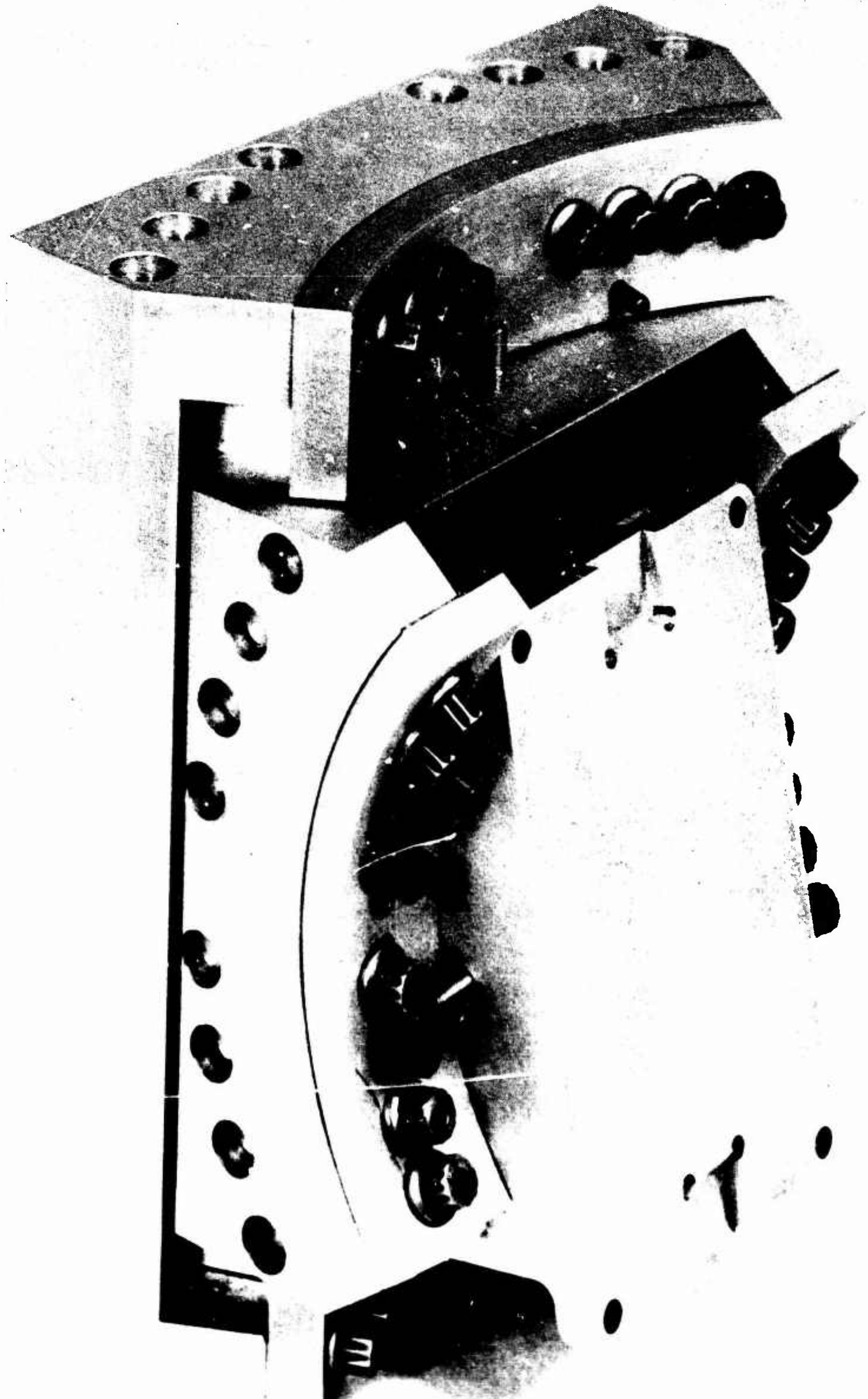


Figure 205. Gimbal Assembly

used extensively in commercial applications such as rod ends, automotive wheel king-pins and aircraft main landing gear trunnion bearings. The space applications include rod ends on the Mariner and the gimbal on the F-1 engine. The nonmetallic surface permits load redistribution which reduces the end loads and notch effects in the proposed application. The 0.012-in. thick PTFE liner is very forgiving and can absorb small dust particles without detrimental effects on gimbal performance. The upper and lower aluminum gimbal blocks have a 0.0016-in. thick hard anodized surface lapped to a 6 to 8 rms finish. Additionally, the PTFE liner will provide a low friction surface between the upper and the lower pivot retainers and the corresponding sides of the upper and lower gimbal blocks.

k. Harnesses

(1) Introduction

Electrical harnesses on the OOS Engine perform the following functions:

1. Interconnection of the engine controller to the valve actuators and igniters.

2. Interconnection between sensors and the engine controller.

Location of the controller at the proposed engine/vehicle interface station 4.50-in. aft of the gimbal point shown on engine assembly Drawing 1161635 eliminates the following interconnect harnesses and resulting additional engine weight:

1. Interconnection of primary power from an interface panel to the engine controller.

2. Interconnection of the vehicle data buses between an interface panel and the engine controller.

Vehicle harnesses with service loops for gimbaling can then be directly connected to the engine mounted controller.

The harnesses are required to perform these functions with extremely high reliability in a severe environment, maintain an adequate degree of isolation for critical low level circuits from externally induced noise, and to have failure modes which do not prevent engine operation. They must offer a high degree of environmental and physical protection to critical circuits carried within and must be producible within the scope of existing materials and fabrication technology for maximum cost effectiveness.

The physical layout and routing of the harnesses would be made such that critical redundant functions are carried in separate harnesses over different routings so that a localized damaging condition does not cause loss of a function. In addition, harnesses would be sized so that replacement of an individual damaged harness may be accomplished in minimum time and at reasonable cost.

(2) Design Description

The harness selected, is a flexible, branching type, environmentally protected by two layers of metallic braid and an abrasion resistant nonmetallic braided covering. Harness assembly and fabrication methods are shown in Figures 206 through 210.

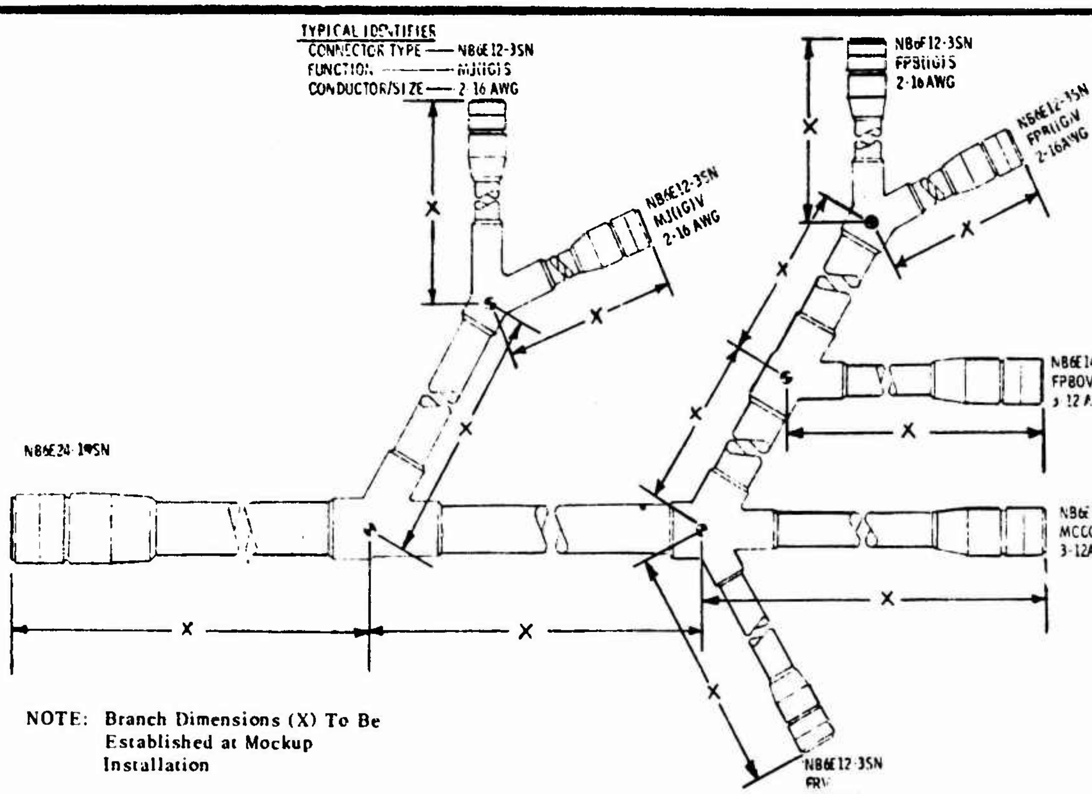


Figure 206. Control Harness Assembly

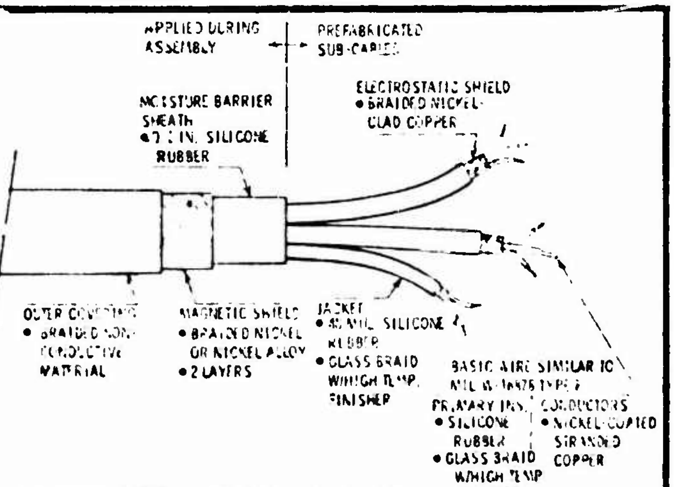


Figure 207. Cable Buildup

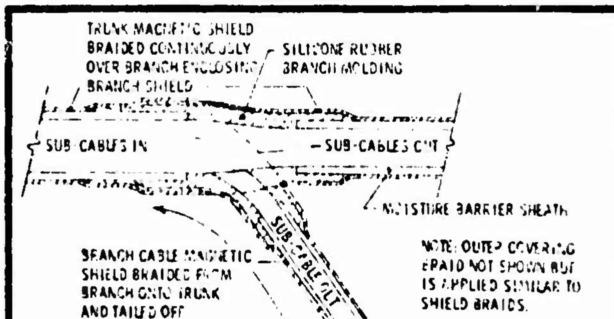


Figure 208. Branch Breakout

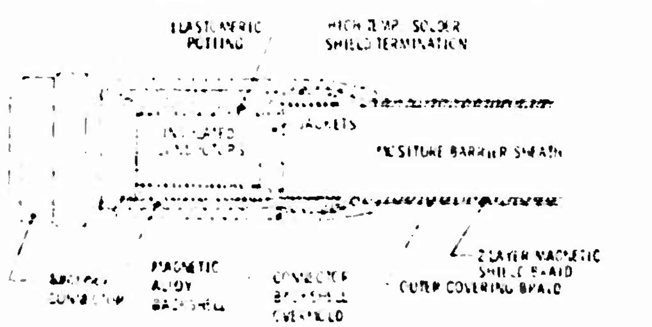


Figure 209. Connector Termination

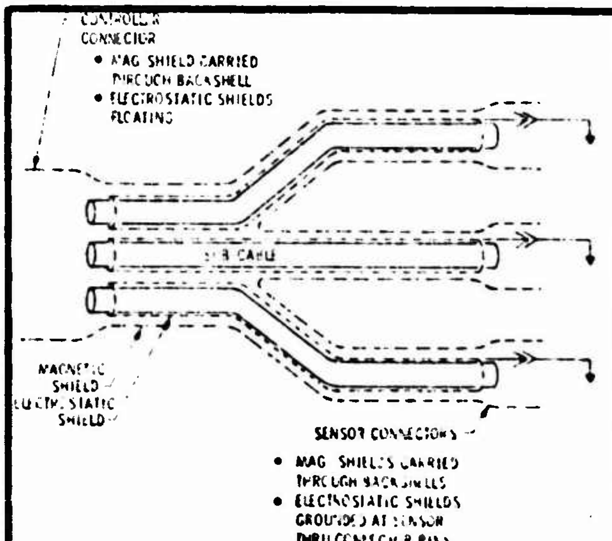


Figure 210. Shield Grounding

The harnesses consist of a main trunk originating at a large connector on the controller routed to the various destinations by means of a number of branches separating from the trunk at appropriate locations. The branches connect to the end devices by suitable smaller connectors.

Basic requirements derived from SSME studies conducted during Phase B along with corresponding capabilities are:

1. Requirement - Adequate insulation and shielding of circuits.

   Capability - Power handling circuits to actuators, solenoids, motors, etc., are carried in harnesses that are separate from the sensors. Each sensor function is connected by a separate electrostatically shielded and jacketed sub-cable within each harness. All harnesses are magnetically shielded.

2. Requirement - Minimum life of 100 hr at 500°R and still function at -100°F.

   Capability - The silicone rubbers specified meet or exceed these requirements.

3. Requirement - Capable of operation after exposure to hydrogen-air flame for three min (aplicable to sea level test firings).

   Capability - Silicone rubber burns to a non-conductive ash which is retained in place by the glass braid providing continuous insulation and preventing engine shutdown.

4. Requirement - Materials and parts compatible with requirements.

   Capability - Harness materials are as follows:

Conductors	- Stranded nickel-clad copper
Dielectrics	- Silicone rubber per MSFC-Dwg-50M60441
Electrostatic Shields	- Braided nickel-clad copper
Outer Covering	- Braided "Nomex" nylon (tentative)
Connectors	- Qualified to MSFC - 40M39569

5. Requirement - Provide 10 hr service-life without overhaul.

   Capability - Heat aging tests of typical electrical silicone rubbers indicate 750 hr at 400°F, 200 hr at 600°F, while retaining adequate physical and electrical properties.

3. Engine Nominal Characteristics Summary

a. Nominal Operating Conditions

(1) Initial Selection of Thrust and Mixture Ratio Control Valve Locations

Initial selection of the steady state control valve locations was performed using the LETS2 model of the baseline 25K engine. Eight possible valve locations were evaluated, as shown schematically in Figure 211 and with the results shown in Table LIV.

Table LIV ranks the valves, in terms of gain, for use as thrust control or mixture ratio control. Selection of the two valves to be used was based on the following criteria:

1. Gain or effectiveness; a high value is desirable for good response.

2. Interaction between controlled variables (amount of thrust variation when controlling mixture ratio or vice versa). A low value (less than $\pm$ 0.2%/%) is desirable.

3. Interaction with remainder of engine system. This was evaluated in Table LIV in terms of preburner gas temperature. A low value is desirable.

4. Valve size. This is represented in Table LIV by the admittance (K_w) of the valve. A low value is desirable.

5. Development risk.

As a first choice, valves giving low interaction and the possibility of a simpler control system are preferred. Of the locations investigated, one thrust control valve location and two mixture ratio control valve locations seem to meet these requirements. They were rejected, however, on other considerations, as follows.

a. Mixture ratio control by a fuel pump recirculation valve. This would recirculate fuel flow from pump discharge back to a location between the high speed inducer and the boost pump turbine. This was rejected on the basis of development risk associated with the problem of mixing the flow. The computer program assumes complete mixing but in actuality, recirculated propellant would first form gas bubbles (since its enthalpy will have been

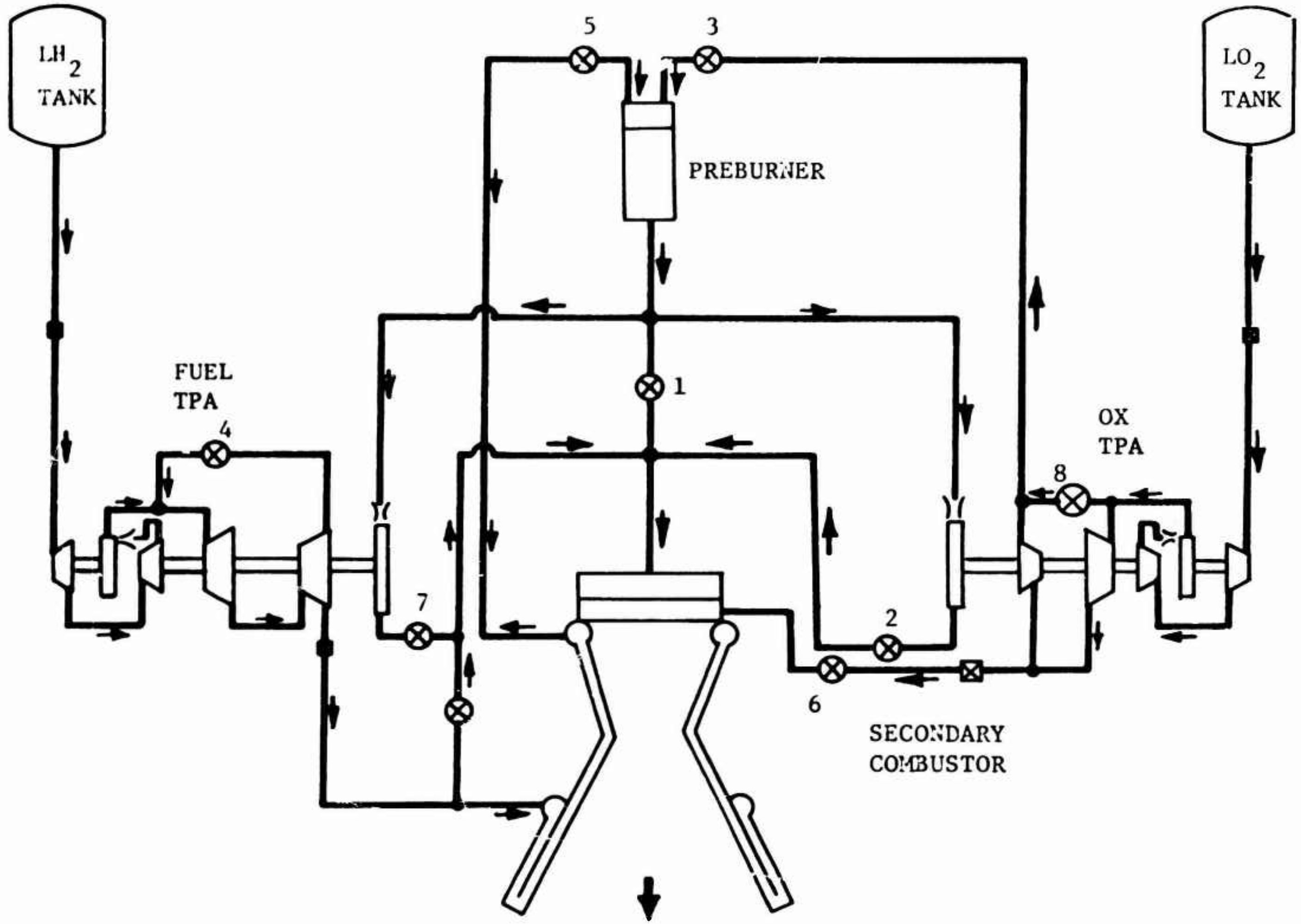


Figure 208. Schematic Candidate Control Valve Locations

TABLE [illegible]

THRUST AND MIXTURE RATIO CONTROL VALVES

THRUST CONTROL VALVES

Location	Nomenclature	$\frac{\Delta F/F}{\Delta X/X}$ (1) %/%	$\frac{\Delta MR/MR}{\Delta F/F}$ %/%	$\frac{\Delta T_G}{100\ F/F}$ °R/%	K_w $\frac{lb\ in.}{sec}$	Fluid Temp °R
1	Turbine Hot Gas Bypass	-1.90	-.175	-2.76	.0555	1860
2	Oxidizer Turbine Exhaust	-0.980	+.662	+13.5	12.7	1743
3*	Oxidizer Preburner Valve	-0.445	+.272	+13.7	1.41	239
4	Fuel Pump Recirculation	-0.250	-6.30	+116	.00455	110
5	Fuel Preburner Inlet	+0.206	+2.36	+49.5	12.2	428
6	Oxidizer Thrust Chamber Inlet	+0.184	-0.230	+17.2	8.55	204
7	Fuel Turbine Exhaust	+0.148	+6.34	+109	27.0	740
8	Oxidizer Pump Recirculation	-0.0825	+0.50	+7.45	.00812	204

MIXTURE RATIO CONTROL VALVES

Location	Nomenclature	$\frac{\Delta MR/MR}{\Delta X/X}$	$\frac{\Delta F/F}{\Delta MR/MR}$	$\frac{\Delta Tr}{100\ \Delta MR/MR}$	K_w $\frac{lb^{\frac{1}{2}}\ in.}{sec}$	Fluid Temp °R
1	Turbine Hot Gas Bypass	+0.331	-5.72	+15.8	.0555	1860
2	Oxidizer Turbine Exhaust	-0.650	+1.51	+20.3	12.7	1743
3	Oxidizer Preburner Inlet	-0.121	+3.68	+50.4	1.41	239
4	Fuel Pump Recirculation	+1.57	-.159	+18.4	.00455	110
5	Fuel Preburner Inlet	+0.471	+.424	+21.0	12.2	428
6*	Oxidizer Thrust Chamber Inlet	-0.042	-4.35	-75.0	8.55	204
7	Fuel Turbine Exhaust	+0.940	+.158	+17.2	27.0	1740
8	Oxidizer Pump Recirculation	-0.041	+2.00	+14.9	.00812	204

(1) NOTE: $\Delta X/X$ is defined as change in pressure drop across valve over steady state pressure drop across valve for direct throttling control valves. (No's. 2, 3, 5, 6, 7)

$\Delta X/X$ is defined as change in flow rate over steady state flow rate for bypass or recirculation control valves. (No's. 1, 4, 8)

* Selected valve location

DATA GENERATED BY
LETS 2 PROGRAM
20 MAY 71 THRU 28 MAY 71

increased by pumping work), which must then collapse before complete mixing could take place.

b. Mixture ratio control by a fuel turbine exhaust throttling valve. This was rejected for three reasons; (1) for use in a two-valve system, the valve must have a nominal pressur-drop sufficient to permit going to -10% mixture ratio on opening the valve. Since it is located in the fuel turbine circuit, this will represent a definite performance penalty in terms of chamber pressure or turbine inlet temperature, (2) it would be the largest valve of any considered, (3) it must operate in high temperature gas.

c. Thrust control by a turbine hot gas bypass valve. This seems an attractive thrust control system, however, when considered in combination with the mixture ratio control systems, it permits wide variations in turbine inlet temperature. For example, the combination of the turbine hot gas bypass valve and an oxidizer turbine exhaust valve will produce a 190° turbine inlet temperature increase at the +10% mixture ratio condition. By comparison, the selected two oxidizer valve system has a 70° variation.

After eliminating the three valve locations discussed above, the remaining locations can be compared. The advantage for hot gas valve location is not considered sufficient to offset their development risk.

The selected system has liquid MR and thrust control valves to minimize the valve development cost, time and weight and consist of the preburner liquid oxidizer valve for thrust control and the liquid oxidizer main injector valve for mixture ratio control. Both control valves are in the oxidizer circuit and therefore have a minimum impact on the nominal power balance.

The selected control method is a two valve control system which was later modified to a three-valve control system for improved off-design and transient engine operations through incorporation of a turbine and gas generator cold gas by-pass valve No. 9.

A further advantage of the selected system is that a preburner oxidizer valve and thrust chamber oxidizer valve will be required for control of engine transient performance in any case.

b. Engine Off-Design Performance Analysis

After selecting the engine baseline configuration and control system, an off-design analysis was conducted. The purpose of this activity is to evaluate individual component performance over the full range of required operating conditions and, if necessary, to modify the design or control system to obtain satisfactory performance.

The analysis was performed by constructing a steady state computer model using the LETS2 system to simulate the engine and its components. Using a specified control system, operating points can be found at the required values of thrust and mixture ratio. These are then examined to determine satisfactory component operation as defined by the following typical parameters:

Pumps - flow coefficient, speed, suction specific speed

Turbines - Inlet temperature, speed

Preburner - Mixture ratio

Injectors - Stiffness, propellant temperature

Cooling Jacket - Flow, pressure drop, bulk temperature rise

Valves - range and pressure drop

The first complete series of off-design runs were made for the two-valve control system selected. This used a preburner oxidizer valve and a main thrust chamber oxidizer valve for thrust and mixture ratio control. Table LV shows operation from 25K to 5K at a constant engine mixture ratio of 6.0 and Table LVI shows operation at 5.5 and 6.5 for thrust levels of 25K, 12.5K and 5K. Selected parameters are also plotted in Figures 212 to 214.

The major problem that became apparent for this two-valve control system was in the preburner mixture ratio and in preburner oxidizer injector stiffness. The overall preburner mixture ratio drops from 0.8 at 25K to 0.5 at 12.5K to a low of 0.26 at 5K, 5.50 MR. To insure reliable combustion, this would require the design of a hot core type of injector and probably some form of mixing device before the turbine inlet. The low mixture ratios also contribute to the second problem, which is the low stiffness in the preburner oxidizer injector. This occurs both because of the low oxidizer flow rates and because the low preburner temperatures do not supply enough heat to completely gasify the incoming oxidizer. This problem could be solved by designing in more pressure drop at the design point. The only penalty would be increased head rise required from the half-stage oxidizer pump. These "fixes" (hot core preburner injector and increased preburner oxidizer injector pressure drop) were not pursued further because the three-valve system developed for a 10:1 throttling range provided a better solution.

Consideration of the requirements of a 10:1 throttling [illegible] also of provision for start and idle mode operation introduced the

TABLE LV

PERFORMANCE DATA, 5:1 THROTTLING AT CONSTANT MIXTURE RATIO

[illegible] ENGINE 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

SUMMARY ENGINE PERFORMANCE DATA

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
ENGINE SPECIFIC IMPULSE	SEC	465.669	465.177	463.285	461.808	460.824	458.623	455.215
TOTAL OXIDIZER FLOW RATE	LB/SEC	46.0212	36.9381	27.8070	23.1881	18.5790	14.0032	9.41290
TOTAL FUEL FLOW RATE	LB/SEC	7.67003	6.15619	4.63485	3.86491	3.09658	2.33340	1.56840
TOTAL FLOW LEAVING SYSTEM	LB/SEC	53.6913	43.0943	32.4418	27.0530	21.6756	16.3366	10.9813
OX TANK OUTFLOW	LB/SEC	46.0209	36.9361	27.8090	23.1889	18.5776	14.0026	9.41285
FUEL TANK OUTFLOW	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
OXIDIZER PRESSURE-INTERFACE	PSIA	46.3584	46.5333	46.6705	46.7252	46.7699	46.8045	46.8294
OXIDIZER ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
OXIDIZER TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902
OXIDIZER DENSITY	LB/FT3	71.0884	71.0881	71.0878	71.0877	71.0876	71.0875	71.0874
FUEL PRESSURE-INTERFACE	PSIA	20.8096	20.9876	21.1273	21.1829	21.2285	21.2637	21.2891
FUEL ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
FUEL TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
FUEL DENSITY	LB/FT3	4.34937	4.34957	4.34973	4.34979	4.34984	4.34988	4.34990
OXID. BOOST PUMP SPEED	RPM	14462.4	12488.2	10591.9	9635.04	8648.27	7609.50	6308.89
OXID. MAIN PUMP SPEED	RPM	49869.7	42441.2	35100.6	31443.1	27634.3	23655.5	18969.0
FUEL BOOST PUMP SPEED	RPM	23150.6	19896.0	16913.0	15467.9	13896.9	12249.9	10178.3
FUEL MAIN PUMP SPEED	RPM	80058.2	67891.3	56320.1	50671.0	44560.3	38204.8	30644.6
OXID PUMP DISCH PRESS-STG1	PSIA	3105.69	2349.44	1695.22	1396.77	1107.03	836.022	559.526
OXID PUMP DISCH PRESS-STG2	PSIA	4665.91	3622.07	2630.95	2162.99	1704.73	1277.77	842.136
FUEL PUMP DISCH PRESS STG3	PSIA	4475.79	3382.82	2421.82	1978.22	1560.11	1157.87	770.941
PRIMARY COMBUSTOR PRESSURE	PSIA	3271.25	2412.04	1678.76	1349.95	1042.86	755.399	483.170
THRUST CHAMBER PRESSURE	PSIA	1800.00	1440.00	1080.00	900.000	720.000	540.000	360.000
OX. TC VALVE ADMITTANCE	KW	1.48840	1.44000	1.24294	1.14113	1.02671	.861998	.804648
OX. PB VALVE ADMITTANCE	KW	.198186	.123102	.791613-01	.616934-01	.477044-01	.339617-01	.202832-01
FU. CJ BYP. VALVE ADMITTANCE	KW							
FU. PB BYP. VALVE ADMITTANCE	KW							

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71 14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71 14:16
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71 14:16
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71 14:18
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71 14:18
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19

NOTE: Boost Pump = LPP

TABLE LV (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
THRUST CHAMBER PERFORMANCE DATA								
THRUST CHAMBER PRESSURE	PSIA	1800.00	1440.00	1080.00	900.000	720.000	540.000	360.000
TC INJECTOR MIXTURE RATIO	-	6.09389	6.09360	6.09261	6.09266	6.09272	6.09367	6.09326
TC AREA RATIO	-	270.000	270.000	270.000	270.000	270.000	270.000	270.000
TC SPECIFIC IMPULSE	SEC	466.263	465.768	463.869	462.389	461.402	459.193	455.773
TC THRUST	LBS	24979.2	20028.0	15015.9	12481.8	9979.40	7485.42	4994.22
DUMP COOLANT THRUST	LBS	23.1278	18.5022	13.8767	11.5639	9.25112	6.93834	4.62556
DUMP COOLANT FLOW RATE	LB/SEC	.117999	.943992-01	.707994-01	.589995-01	.471996-01	.353997-01	.235998-01
REGEN COOLANT FLOW RATE	LB/SEC	7.33920	5.89803	4.44198	3.70270	2.97246	2.23796	1.50790
INLET PRESSURE	PSIA	4417.02	3344.91	2400.19	1963.17	1550.32	1152.27	768.403
INLET TEMPERATURE	R	121.592	101.600	85.0914	78.1350	71.5490	65.0826	57.8414
INLET DENSITY	LB/FT3	4.06291	4.07370	4.05521	4.05194	4.01424	3.98632	3.99077
INLET ENTHALPY	BTU/LB	238.928	153.517	83.3617	53.0524	24.2770	-2.91554	-32.3148
PRESSURE DROP	PSI	858.942	693.845	531.447	448.992	369.256	286.118	203.214
ENTHALPY RISE	BTU/LB	1279.33	1332.11	1406.73	1459.53	1523.73	1612.88	1740.61
TEMPERATURE RISE	R	324.935	340.361	361.000	375.164	392.527	417.204	452.730
OUTLET PRESSURE	PSIA	3558.08	2651.07	1868.75	1514.18	1181.06	866.153	565.189
OUTLET TEMPERATURE	R	446.527	441.961	446.091	453.299	464.076	482.286	510.572
OUTLET DENSITY	LB/FT3	1.27556	.998784	.723063	.585085	.454316	.324786	.203504
OUTLET ENTHALPY	BTU/LB	1518.25	1485.62	1490.09	1512.58	1548.00	1609.97	1708.30
GAS INJECTOR FLOW RATE	LB/SEC	13.5627	10.0753	7.01067	5.61698	4.32267	3.10868	1.95644
INLET PRESSURE	PSIA	1993.88	1575.13	1168.58	968.926	771.371	575.664	381.726
PRESSURE DROP	PSI	139.885	91.9272	56.1800	41.9264	29.7708	19.4644	10.9258
ΔP/PCFACE	-	.754503-01	.619789-01	.505034-01	.452280-01	.401440-01	.349954-01	.294654-01
TC OXID. INJECTOR FLOW RATE	LB/SEC	39.9087	32.8424	25.3042	21.3295	17.2653	13.1627	8.98257
INLET PRESSURE	PSIA	2466.69	1887.29	1327.63	1086.84	855.457	629.539	448.802
INLET TEMPERATURE	R	184.156	179.065	174.931	172.958	170.590	169.901	167.728
INLET ENTHALPY	BTU/LB	-43.7952	-46.9534	-49.6058	-50.7778	-51.8869	-52.9339	-54.0693
OUTLET TEMPERATURE	R	552.825	462.079	383.097	350.530	322.396	300.901	260.257
OUTLET DENSITY	LB/FT3	11.1786	12.0736	11.7006	11.2071	10.2929	7.94709	6.62923
PRESSURE DROP	PSI	612.692	404.086	215.226	159.841	113.857	73.3390	78.0025
ΔP/PCFACE	-	.330470	.272442	.193479	.172428	.153529	.131857	.210363
TC IGNITER OXID. FLOW RATE	LB/SEC	.688548-01	.514396-01	.370323-01	.301398-01	.243250-01	.188209-01	.121053-01
FUEL FLOW RATE	LB/SEC	.101262	.796241-01	.587518-01	.486516-01	.386771-01	.289161-01	.184519-01
MIXTURE RATIO	-	.679965	.646030	.630317	.619503	.628925	.650881	.656045

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
PREBURNER PERFORMANCE DATA								
PREBURNER PRESSURE	PSIA	3271.25	2412.04	1678.76	1349.95	1042.86	755.399	483.170
MIXTURE RATIO	-	.811086	.675723	.547750	.486887	.427798	.361802	.273871
GAS TEMPERATURE	R	1878.82	1646.13	1432.44	1332.08	1229.75	1125.57	998.154
SPECIFIC HT. RATIO	-	1.36418	1.37149	1.37781	1.38067	1.38349	1.38648	1.39021
GAS CONSTANT	FT	422.473	456.884	495.053	515.556	537.167	563.567	603.088
FLOW RATE	LB/SEC	13.4942	10.0242	6.97304	5.58648	4.29822	3.09054	1.94489
PB OXID. INJECTOR FLOW RATE	LB/SEC	5.95248	3.97022	2.41006	1.78037	1.24984	.791606	.398419
INLET PRESSURE	PSIA	3847.98	2683.28	1795.25	1411.56	1085.61	785.811	494.586
INLET TEMPERATURE	R	206.445	197.138	189.816	186.733	183.763	180.141	177.520
INLET DENSITY	LB/FT3	67.4570	67.6919	67.7778	67.9524	68.1992	67.5848	68.7369
TEMPERATURE RISE	R	205.124	181.973	153.918	140.316	125.500	126.781	144.638
ENTHALPY RISE	BTU/LB	83.7229	81.1893	78.8795	78.3649	78.7704	93.0049	108.534
OUTLET TEMPERATURE	R	411.569	379.111	343.734	327.049	309.264	306.922	322.157
OUTLET DENSITY	LB/FT3	30.6872	28.4574	25.2636	23.1067	20.1781	11.3207	5.66131
PRESSURE DROP	PSI	576.726	271.237	116.492	61.6084	42.7535	30.4123	11.4163
ΔP/PCPB	-	.176301	.112451	.693914-01	.456377-01	.409965-01	.402600-01	.236280-01
PB FUEL INJECTOR FLOW RATE	LB/SEC	7.33825	5.89534	4.44207	3.70046	2.97601	2.23808	1.50872
INLET PRESSURE	PSIA	3538.27	2634.73	1855.95	1503.19	1171.94	858.920	559.949
INLET TEMPERATURE	R	446.527	441.961	446.091	453.299	464.076	482.286	510.572
INLET DENSITY	LB/FT3	1.26919	.993125	.718359	.581095	.450887	.322155	.201627
INLET ENTHALPY	BTU/LB	1518.25	1485.62	1490.09	1512.58	1548.00	1609.97	1708.30
PRESSURE DROP	PSI	267.024	222.690	177.187	153.241	129.083	103.521	76.7790
ΔP/PCPB	-	.816275-01	.923243-01	.105546	.113516	.123778	.137042	.158907
PB IGNITER OXID. FLOW RATE	LB/SEC	.908130-01	.722177-01	.575368-01	.487557-01	.379826-01	.289911-01	.196862-01
FUEL FLOW RATE	LB/SEC	.111696	.838207-01	.632818-01	.544716-01	.379210-01	.314822-01	.188582-01
MIXTURE RATIO	-	.813034	.861574	.909217	.895067	1.00163	.920875	1.04391

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   5:1 THROTTLEABLE   THROTTLED AT CONSTANT MIXTURE RATIO   PAGE 4

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
FUEL BOOST PUMP AND HYDRAULIC TURBINE								
LOW SPEED FUEL INDUCER								
SUCTION PRESSURE	PSIA	19.3069	20.0197	20.5786	20.8015	20.9836	21.1246	21.2262
ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
TEMPERATURE	R	37.1297	37.6857	37.7954	37.7934	37.7919	37.7906	37.7898
SHAFT SPEED	RPM	23150.6	19896.0	16913.0	15467.9	13896.9	12249.9	10178.3
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
VOLUME FLOW RATE	GPM	791.766	635.351	478.284	398.801	319.481	240.796	161.864
HEAD RISE (ACTUAL)	FT	906.165	719.741	575.569	512.261	441.356	367.080	272.403
PRESSURE RISE	PSI	28.5951	22.6628	18.0927	16.0906	13.8514	11.5097	8.53070
(Q/N)/(QD/ND)	-	.987155	.921718	.816235	.744176	.663554	.567370	.459016
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3944.17	3609.03	3147.68	2868.74	2579.58	2266.67	1931.25
EFFICIENCY	-	.709794	.699572	.681966	.663040	.634917	.589292	.521732
NPSH (NO TSH)	FT	40.7396	65.1842	84.6456	92.1900	98.3561	103.127	106.567
NPSP (NO TSH)	PSI	1.23003	1.96842	2.55648	2.78450	2.97088	3.11510	3.21909
THERMODYNAMIC SUCT. HEAD	FT	120.569	126.096	127.186	127.167	127.151	127.139	127.130
SUCTION SPECIFIC SPEED		14391.9	9750.37	6661.48	5419.34	4268.46	3215.77	2166.50
S/SDESIGN	-	.319819	.216675	.148033	.120430	.948546-01	.714616-01	.481444-01
S/SMAX AT Q/QD	-	.323109	.219125	.156549	.136297	.121273	.114102	.102083
SHAFT HORSEPOWER	HP	17.8040	11.5154	7.11223	5.42898	3.91334	2.64317	1.48926
DISCHARGE PRESSURE	PSIA	28.5951	22.6628	18.0927	16.0906	13.8514	11.5097	8.53070
ENTHALPY	BTU/LB	-104.619	-104.938	-105.175	-105.267	-105.367	-105.460	-105.589
TEMPERATURE	R	38.1035	38.0411	37.9968	37.9828	37.9694	37.9607	37.9462
HYDRAULIC TURBINE								
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
INLET PRESSURE	PSIA	198.118	160.717	129.250	113.695	97.4761	81.5105	63.2324
ENTHALPY	BTU/LB	-96.0091	-98.2772	-100.054	-100.794	-101.571	-102.308	-103.197
TEMPERATURE	R	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337
PRESSURE DROP	PSI	38.7681	31.1988	25.5766	23.3799	21.0196	18.8177	15.7449
ISEN. SPOUTING VELOCITY	FT/SEC	328.092	294.175	267.014	256.390	245.000	235.201	221.045
MEAN WHEEL SPEED	FT/SEC	205.725	176.803	150.295	137.454	123.493	108.857	90.4477
VELOCITY RATIO - U/CO	-	.627031	.601012	.562872	.536110	.504053	.462825	.409181
EFFICIENCY	-	.763447	.764998	.762071	.756326	.745437	.725014	.687625
SHAFT HORSEPOWER	HP	17.8106	11.5153	7.11541	5.42929	3.91456	2.64475	1.48931
OUTLET PRESSURE	PSIA	159.350	129.518	103.673	90.3149	76.4565	62.6928	47.4875
ENTHALPY	BTU/LB	-97.6503	-99.5993	-101.139	-101.787	-102.465	-103.109	-103.868
TEMPERATURE	R	39.7519	39.2282	38.8661	38.7506	38.6342	38.5360	38.4145
TURBINE TORQUE	FT-LBS	4.04063	3.03978	2.20960	1.84351	1.47944	1.13393	.768503

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20	MRE=6.00	15 JUL 71 14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71 14:16
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71 14:16
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71 14:18
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71 14:18
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19

TABLE LV (cont.)

[illegible] [illegible] THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
FUEL PUMP - HIGH SPEED INDUCER AND FIRST STAGE								
HIGH SPEED FUEL INDUCER								
SUCTION PRESSURE	PSIA	47.9020	42.6825	38.6714	36.8921	34.8350	32.6343	29.7569
ENTHALPY	BTU/LB	-104.619	-104.938	-105.175	-105.267	-105.367	-105.460	-105.589
TEMPERATURE	R	38.0924	38.0342	37.9930	37.9802	37.9677	37.9598	37.9458
SHAFT SPEED	RPM	80058.2	67891.3	56320.1	50671.0	44560.3	38204.8	30644.6
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
VOLUME FLOW RATE	GPM	791.302	635.120	478.206	398.791	319.526	240.882	161.959
HEAD RISE (ACTUAL)	FT	4761.13	3626.76	2743.84	2343.77	1926.54	1514.90	1046.74
PRESSURE RISE	PSI	150.216	118.034	90.5782	76.8027	62.6411	48.8762	33.4755
(Q/N)/(QD/ND)	-	.998392	.944945	.857663	.794970	.724307	.636872	.533847
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3916.78	3661.03	3248.64	3003.97	2739.16	2441.93	2119.22
EFFICIENCY	-	.713557	.699704	.688520	.673367	.652247	.617716	.562396
NPSH (NO TSH)	FT	961.897	794.595	665.672	607.998	541.102	469.050	375.117
NPSP (NO TSH)	PSI	29.0590	24.0038	20.1080	18.3644	16.3419	14.1632	11.3245
THERMODYNAMIC SUCT. HEAD	FT	130.132	129.560	129.150	129.023	128.899	128.820	128.681
SUCTION SPECIFIC SPEED		11854.9	10207.8	8227.51	7153.56	6048.45	4904.16	3667.46
S/SDESIGN	-	.395165	.340261	.274250	.238452	.201615	.163472	.122249
S/SMAX AT Q/QD	-	.399893	.343249	.283004	.256362	.233983	.220367	.214598
SHAFT HORSEPOWER	HP	93.4423	58.0151	33.5825	24.4585	16.6280	10.4061	5.30891
DISCHARGE PRESSURE	PSIA	198.118	160.717	129.250	113.695	97.4761	81.5105	63.2324
ENTHALPY	BTU/LB	-96.0091	-98.2772	-100.054	-100.794	-101.571	-102.308	-103.197
TEMPERATURE	R	40.0816	39.4606	39.0347	38.8959	38.7565	38.6383	38.4930
FUEL PUMP - STAGE 1								
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
VOLUME FLOW RATE (1)	GPM	921.564	726.277	542.409	452.576	363.403	275.820	187.762
HEAD RISE (ACTUAL)	FT	49605.1	37005.5	26332.5	21593.8	16908.7	12614.8	8183.15
PRESSURE RISE	PSI	1466.79	1096.05	788.629	645.153	504.355	378.422	240.619
(Q/N)/(QD/ND)		1.00644	.935313	.842039	.780910	.713034	.631215	.535700
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	731.179	685.751	634.538	605.145	572.874	533.055	486.735
EFFICIENCY	-	.606118	.599161	.585128	.570908	.545971	.516530	.469699
NPSH (NO TSH)	FT	3559.15	2885.54	2216.46	1833.13	1432.24	1074.25	677.984
NPSP (NO TSH)	PSI	98.7807	81.8607	63.8336	53.0748	41.6998	31.4359	19.9658
THERMODYNAMIC SUCT. HEAD	FT	219.274	198.160	182.805	177.027	171.180	163.759	155.103
SUCTION SPECIFIC SPEED		5042.96	4421.41	3826.20	3590.72	3352.40	3040.10	2707.95
S/SDESIGN	-	.504296	.442141	.382620	.359072	.335240	.304010	.270795
S/SMAX AT Q/QD	-	.507998	.432293	.400858	.402275	.414096	.435922	.462467
SHAFT HORSEPOWER	HP	1221.14	742.368	410.114	289.131	191.159	114.994	56.1983
DISCHARGE PRESSURE	PSIA	1626.14	1225.57	892.302	735.468	580.811	441.115	288.106
ENTHALPY	BTU/LB	28.2682	-4.52366	-31.4805	-42.9131	-54.0176	-64.5798	-76.0970
TEMPERATURE	R	72.3663	64.1458	57.4443	54.6212	52.2956	49.3911	46.7897

(1) VOLUME FLOW INCLUDES 7% RECIRCULATION

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71 14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71 14:16
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71 14:16
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71 14:18
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71 14:18
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19

TABLE LV (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
FUEL PUMP - SECOND AND THIRD STAGES								
SECOND STAGE FUEL PUMP								
SUCTION PRESSURE	PSIA	1626.14	1225.57	892.302	735.468	580.811	441.115	288.106
ENTHALPY	BTU/LB	28.2682	-4.52366	-31.4805	-42.9131	-54.0176	-64.5798	-76.0970
TEMPERATURE	R	72.3547	64.1316	57.4359	54.6161	52.2904	49.3891	46.7884
SHAFT SPEED	RPM	80058.2	67891.3	56320.1	50671.0	44560.3	38204.8	30644.6
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
VOLUME FLOW RATE (1)	GPM	913.890	728.618	547.398	459.529	370.256	281.832	192.290
HEAD RISE (ACTUAL)	FT	49760.6	36936.9	26270.0	21542.5	16864.8	12582.0	8203.23
PRESSURE RISE	PSI	1449.36	1084.84	769.139	625.718	493.053	360.950	244.635
(Q/N)/(QD/ND)	-	.999153	.939354	.850714	.793775	.727274	.645678	.549219
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	726.422	687.811	638.587	610.863	579.378	539.884	492.996
EFFICIENCY	-	.607008	.599323	.585917	.574734	.550102	.520694	.477641
SHAFT HORSEPOWER	HP	1223.18	740.791	408.589	286.525	189.231	113.778	55.3995
DISCHARGE PRESSURE	PSIA	3075.50	2310.41	1651.44	1361.19	1073.86	802.064	532.741
ENTHALPY	BTU/LB	133.612	74.6748	26.1353	5.25358	-14.6212	-33.5281	-54.0271
TEMPERATURE	R	97.2033	83.0454	71.4587	66.5699	62.0156	57.3125	52.4831
THIRD STAGE FUEL PUMP								
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63483	3.86482	3.09626	2.33377	1.56881
VOLUME FLOW RATE (1)	GPM	905.296	727.947	551.763	463.538	375.657	287.503	195.950
HEAD RISE (ACTUAL)	FT	49747.2	36789.3	26150.6	21481.7	16809.5	12537.7	8191.14
PRESSURE RISE	PSI	1400.29	1072.42	760.385	617.032	486.251	355.801	238.200
(Q/N)/(QD/ND)	-	.999600	.947821	.866025	.808662	.745221	.665220	.565241
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	723.144	689.561	643.321	614.824	585.028	546.732	498.218
EFFICIENCY	-	.607005	.599627	.587223	.577522	.555322	.526304	.484793
SHAFT HORSEPOWER	HP	1222.85	737.459	405.828	284.336	186.838	112.170	54.5016
FUEL TURBINE SPEED	RPM	80058.2	67891.3	56320.1	50671.0	44560.3	38204.8	30644.6
COMBINED FUEL PUMP PERFORMANCE BASED ON DELIVERED FLOW								
TOTAL FUEL TPA POWER REQ'D	HP	3760.62	2278.63	1258.11	884.450	583.855	351.348	171.408
TOTAL FUEL TPA HEAD RISE	FT	153894.	114358.	81496.9	66961.7	52509.6	39249.4	25624.3
OVERALL PUMP EFFICIENCY	-	.570695	.561734	.545874	.532010	.506301	.474012	.426409

(1) VOL. FLOW INCLUDES 7% RECIRC.

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
OXIDIZER BOOST PUMP AND HYDRAULIC TURBINE								
LOW SPEED OXIDIZER INDUCER								
SUCTION PRESSURE	PSIA	44.8915	45.5884	46.1349	46.3528	46.5309	46.6687	46.7681
ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
TEMPERATURE	R	162.927	162.922	162.918	162.915	162.910	162.907	162.904
SHAFT SPEED	RPM	14462.4	12488.2	10591.9	9635.04	8648.27	7609.50	6308.89
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8090	23.1889	18.5776	14.0026	9.41285
VOLUME FLOW RATE	GPM	290.529	233.182	175.564	146.398	117.286	88.4029	59.4265
HEAD RISE (ACTUAL)	FT	246.882	198.859	157.898	138.550	119.202	98.7796	72.9974
PRESSURE RISE	PSI	121.878	98.1704	77.9489	68.3974	58.8458	48.7638	36.0361
(Q/N)/(QD/ND)	-	.969884	.920090	.816765	.748712	.668266	.572459	.464153
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3957.92	3601.11	3150.70	2886.79	2596.20	2283.44	1947.43
EFFICIENCY	-	.660592	.649962	.632667	.615393	.589288	.547135	.484478
NPSH (NO TSH)	FT	60.2236	61.6441	62.7580	63.2051	63.5729	63.8575	64.0628
NPSP (NO TSH)	PSI	29.7319	30.4325	30.9819	31.2024	31.3838	31.5242	31.6255
THERMODYNAMIC SUPT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		10865.9	8268.93	6009.16	4966.71	3973.95	3026.15	2052.39
S/SDESIGN	-	.241464	.183754	.133537	.110371	.883100-01	.672479-01	.456088-01
S/SMAX AT Q/QD	-	.244034	.185881	.141166	.124251	.111931	.105900	.958714-01
SHAFT HORSEPOWER	HP	31.2715	20.5469	12.6190	9.49234	6.83255	4.59641	2.57865
DISCHARGE PRESSURE	PSIA	121.878	98.1704	77.9489	68.3974	58.8458	48.7638	36.0361
ENTHALPY	BTU/LB	-56.5647	-56.6518	-56.7243	-56.7557	-56.7851	-56.8130	-56.8514
TEMPERATURE	R	163.119	163.068	163.038	163.025	163.088	163.049	163.017
HYDRAULIC TURBINE								
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8090	23.1889	18.5776	14.0026	9.41285
INLET PRESSURE	PSIA	785.917	617.631	480.490	415.843	352.958	289.717	217.130
ENTHALPY	BTU/LB	-54.1338	-54.7462	-55.2673	-55.4974	-55.7170	-55.9283	-56.1791
TEMPERATURE	R	165.309	165.309	165.309	165.309	165.309	165.309	165.309
PRESSURE DROP	PSI	184.164	150.238	122.600	110.602	99.3045	88.6471	74.0127
ISEN. SPOUTING VELOCITY	FT/SEC	177.347	160.546	145.619	139.029	133.044	127.705	120.263
MEAN WHEEL SPEED	FT/SEC	107.233	92.5949	78.5346	71.4401	64.1236	56.4215	46.7780
VELOCITY RATIO - U/CO	-	.604651	.576750	.539314	.513851	.481973	.441809	.388963
EFFICIENCY	-	.764954	.763851	.757174	.749229	.735398	.711823	.670360
SHAFT HORSEPOWER	HP	31.2852	20.5475	12.6160	9.48867	6.83284	4.59305	2.57867
OUTLET PRESSURE	PSIA	601.752	467.393	357.891	305.241	253.654	201.070	143.118
ENTHALPY	BTU/LB	-54.6143	-55.1394	-55.5879	-55.7866	-55.9770	-56.1601	-56.3727
TEMPERATURE	R	166.225	165.565	164.678	164.589	164.189	163.867	163.749
TURBINE TORQUE	FT-LBS	11.3614	8.64161	6.25578	5.17232	4.14960	3.17014	2.14673

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71 14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71 14:16
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71 14:16
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71 14:18
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71 14:18
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19

TABLE LV (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
OXIDIZER PUMP - HIGH SPEED INDUCER AND MAIN STAGE								
HIGH SPEED OXIDIZER INDUCER								
SUCTION PRESSURE	PSIA	166.770	143.759	124.084	114.750	105.377	95.4325	82.8042
ENTHALPY	BTU/LB	-56.5647	-56.6518	-56.7243	-56.7557	-56.7851	-56.8130	-56.8514
TEMPERATURE	R	163.120	163.085	163.031	163.039	163.097	163.044	163.019
SHAFT SPEED	RPM	49869.7	42441.2	35100.6	31443.1	27634.3	23655.5	18969.0
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8090	23.1889	18.5776	14.0026	9.41285
VOLUME FLOW RATE	GPM	291.432	233.813	175.984	146.730	117.466	88.5515	59.5322
HEAD RISE (ACTUAL)	FT	1258.01	962.472	723.673	611.292	502.286	394.216	272.585
PRESSURE RISE	PSI	619.147	473.873	356.406	301.093	247.582	194.284	134.326
(Q/N)/(QD/ND)	-	1.00756	.949845	.864429	.804573	.732883	.645409	.541102
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4030.34	3755.61	3337.29	3098.12	2822.87	2516.11	2181.70
EFFICIENCY	-	.665005	.649040	.638264	.624293	.604342	.572609	.521027
NPSH (NO TSH)	FT	307.759	260.958	221.012	202.024	182.774	162.703	137.131
NPSP (NO TSH)	PSI	151.467	128.483	108.848	99.5075	90.0912	80.1862	67.5765
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		11474.7	9881.86	8014.88	7003.99	5928.12	4798.14	3574.40
S/SDESIGN	-	.382490	.329395	.267163	.233466	.197604	.159938	.119147
S/SMAX AT Q/QD	-	.400133	.332214	.274835	.249054	.226771	.211826	.204827
SHAFT HORSEPOWER	HP	158.289	99.5876	57.3277	41.2836	28.0733	17.5276	8.95363
DISCHARGE PRESSURE	PSIA	785.917	617.631	480.490	415.843	352.958	289.717	217.130
ENTHALPY	BTU/LB	-54.1338	-54.7462	-55.2673	-55.4974	-55.7170	-55.9283	-56.1791
TEMPERATURE	R	167.010	165.841	165.150	164.854	164.395	164.321	163.717
OXIDIZER PUMP - STAGE 1								
WT. FLOW RATE (DELIVERED)	LB/SEC	46.0209	36.9361	27.8090	23.1889	18.5776	14.0026	9.41285
VOLUME FLOW RATE (1)	GPM	307.888	247.385	187.254	156.582	126.141	95.8787	65.2751
HEAD RISE (ACTUAL)	FT	5105.12	3833.07	2726.27	2222.91	1737.93	1292.99	847.239
PRESSURE RISE	PSI	2503.94	1882.04	1337.33	1091.53	853.380	634.952	416.408
(Q/N)/(QD/ND)		1.01544	.958699	.877432	.819056	.750765	.666631	.565978
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	1448.87	1370.30	1273.08	1215.36	1153.06	1074.23	975.919
EFFICIENCY	-	.635209	.630701	.614533	.599192	.574623	.543276	.500242
NPSH (NO TSH)	FT	1192.19	917.402	696.023	588.069	483.614	376.929	258.832
NPSP (NO TSH)	PSI	583.075	449.466	340.862	288.359	237.173	184.901	127.111
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		4302.12	3991.51	3529.36	3278.06	2991.02	2686.34	2347.80
S/SDESIGN	-	.430212	.399151	.352936	.327806	.299102	.268634	.234780
S/SMAX AT Q/QD	-	.438299	.387457	.358615	.351426	.348849	.359818	.382239
SHAFT HORSEPOWER	HP	706.006	429.715	237.332	166.169	109.149	65.3014	31.6727
DISCHARGE PRESSURE	PSIA	3105.69	2349.44	1695.22	1396.77	1107.03	836.022	559.526
ENTHALPY	BTU/LB	-43.7952	-46.9534	-49.6058	-50.7778	-51.8869	-52.9339	-54.0693
TEMPERATURE	R	184.158	179.065	174.931	172.958	170.590	169.901	167.728

(1) VOLUME FLOW INCLUDES 5% RECIRCULATION.

CASE 1 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2 LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3 LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5 LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6 LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

[illegible] 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO PAGE 9

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
OXIDIZER PUMP - MAIN STAGE								
SUCTION PRESSURE	PSIA	2846.36	2233.42	1652.01	1373.01	1095.22	831.236	558.280
ENTHALPY	BTU/LB	-43.7952	-46.9534	-49.6058	-50.7778	-51.8869	-52.9339	-54.0693
TEMPERATURE	R	186.539	180.714	176.178	173.988	171.518	170.813	168.566
SHAFT SPEED	RPM	49869.7	42441.2	35100.6	31443.1	27634.3	23655.5	18969.0
WEIGHT FLOW RATE	LB/SEC	6.04331	4.04217	2.46777	1.82932	1.28783	.821091	.418135
VOLUME FLOW RATE	GPM	40.8821	27.6390	17.2193	13.0008	9.41227	6.22787	3.47391
HEAD RISE (ACTUAL)	FT	3713.54	2833.99	1996.79	1612.55	1248.41	911.013	581.162
PRESSURE RISE	PSI	1819.55	1388.65	978.948	789.983	609.511	446.534	283.855
(Q/N)/(QO/NO)	-	.931567	.740036	.557465	.469855	.387046	.299174	.208109
SPECIFIC SPEED RPM-GPM**.5/FT**.75		670.282	574.447	487.610	445.529	403.672	356.008	298.697
EFFICIENCY	-	.454885	.426588	.374173	.337007	.296214	.250105	.192281
SHAFT HORSEPOWER	HP	94.5882	52.2082	26.1921	17.7202	11.2660	6.47188	2.98581
DISCHARGE PRESSURE	PSIA	4665.91	3622.07	2630.95	2162.99	1704.73	1277.77	842.136
ENTHALPY	BTU/LB	-32.7854	-37.9448	-42.3209	-44.2221	-46.0883	-47.8919	-49.8503
TEMPERATURE	R	206.445	197.138	189.816	186.733	183.763	180.141	177.520

CASE 1	LETS2 0.5 STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 0.5 STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 0.5 STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 0.5 STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 0.5 STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 0.5 STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 0.5 STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
GAS TURBINE PERFORMANCE DATA								
FUEL TPA SHAFT SPEED	RPM	80058.2	67891.3	56320.1	50671.0	44560.3	38204.8	30644.6
TURBINE INLET TEMPERATURE	R	1860.66	1631.30	1420.69	1321.69	1220.55	1116.40	990.101
WEIGHT FLOW RATE (1)	LB/SEC	10.0313	7.45293	5.18400	4.15327	3.19540	2.29733	1.44595
MEAN WHEEL SPEED	FT/SEC	1299.89	1102.34	914.460	822.737	723.518	620.325	497.572
ISEN. SPOUTING VELOCITY	FT/SEC	4763.78	4317.63	3879.97	3660.13	3429.37	3194.41	2904.36
STAGE VEL. RATIO U/C	-	.365895	.361064	.333311	.317891	.298366	.274626	.242281
INLET PRESSURE (TOTAL)	PSIA	3215.41	2375.15	1656.30	1333.23	1031.00	747.597	478.805
EXIT PRESSURE (STATIC)	PSIA	1993.88	1575.13	1168.58	968.926	771.371	575.664	381.726
PRESSURE RATIO	-	1.61264	1.50791	1.41736	1.37599	1.33658	1.29867	1.25432
SHAFT HORSEPOWER	HP	3760.87	2279.14	1258.07	884.268	583.694	351.292	171.348
SHAFT TORQUE	FT-LBS	246.727	176.316	117.321	91.6555	68.7974	48.2931	29.3669
EXIT TEMPERATURE	R	1730.36	1531.58	1346.75	1259.06	1168.71	1074.80	959.767
EFFICIENCY	-	.628503	.624107	.613322	.604654	.590934	.570125	.534479
OXID TPA SHAFT SPEED	RPM	49869.7	42441.2	35100.6	31443.1	27634.3	23655.5	18969.0
TURBINE INLET TEMPERATURE	R	1860.66	1631.30	1420.69	1321.69	1220.55	1116.40	990.101
WEIGHT FLOW RATE (1)	LB/SEC	3.46200	2.57202	1.78900	1.43330	1.10274	.792811	.498939
MEAN WHEEL SPEED	FT/SEC	1096.26	932.962	771.598	691.197	607.471	520.007	416.986
ISEN. SPOUTING VELOCITY	FT/SEC	4763.78	4317.63	3879.97	3660.13	3429.37	3194.41	2904.36
VELOCITY RATIO - U/C	-	.230123	.216081	.198867	.188844	.177138	.162786	.143572
INLET PRESSURE (TOTAL)	PSIA	3215.41	2375.15	1656.30	1333.23	1031.00	747.597	478.805
EXIT PRESSURE (STATIC)	PSIA	1993.88	1575.13	1168.58	968.926	771.371	575.664	381.726
PRESSURE RATIO	-	1.61264	1.50791	1.41736	1.37599	1.33658	1.29867	1.25432
SHAFT HORSEPOWER	HP	959.195	581.594	320.890	225.226	148.593	89.3318	43.5412
SHAFT TORQUE	FT-LBS	101.019	71.9726	48.0149	37.6208	28.2412	19.8339	12.0556
EXIT TEMPERATURE	R	1764.36	1557.57	1366.04	1275.47	1182.31	1085.75	967.765
EFFICIENCY	-	.464497	.461489	.453309	.446268	.435918	.420108	.393556
TURBINE GAS PROPERTIES								
SPECIFIC HEAT RATIO	-	1.36413	1.37149	1.37781	1.38067	1.38349	1.38648	1.39021
GAS CONSTANT	FT	422.473	456.884	495.053	515.556	537.167	563.567	603.088
MOLECULAR WEIGHT	-	3.6546	3.37942	3.11886	2.99483	2.87434	2.73969	2.56016
TEMPERATURE	R	1860.66	1631.30	1420.69	1321.69	1220.55	1116.40	990.101

(1) INCLUDES 7% LEAKAGE FLOW

CASE 1	LETS2 O S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 O S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 O S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 O S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 O S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 O S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 O S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

EARTH TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
CONTROLS COMPONENTS								
OXID. THRUST CHAMBER VALVE								
WEIGHT FLOW RATE	LB/SEC	39.9084	32.8425	25.3042	21.3295	17.2654	13.1627	8.98261
INLET PRESSURE	PSIA	3105.69	2349.44	1695.22	1396.77	1107.03	836.022	559.526
INLET TEMPERATURE	R	184.158	179.065	174.931	172.958	170.590	169.901	167.728
INLET DENSITY	LB/FT3	70.5557	70.5598	70.5974	70.5451	70.3051	70.5819	70.3335
PRESSURE DROP	PSI	635.843	460.017	366.336	309.033	250.987	206.143	110.564
ADMITTANCE	KW	1.48840	1.44000	1.24294	1.14113	1.02671	.861998	.804648
OXID. PREBURNER VALVE								
WEIGHT FLOW RATE	LB/SEC	5.95250	3.96995	2.41023	1.78056	1.24985	.792100	.398449
INLET PRESSURE	PSIA	4665.91	3622.07	2630.95	2162.99	1704.73	1277.77	842.136
INLET TEMPERATURE	R	206.445	197.138	189.816	186.733	183.763	180.141	177.520
INLET DENSITY	LB/FT3	68.8206	69.1277	69.2188	69.1719	69.1852	68.9981	69.2848
PRESSURE DROP	PSI	817.932	938.794	835.700	751.436	619.116	491.959	347.549
ADMITTANCE	KW	.198186	.123102	.791613-01	.616934-01	.477044-01	.339617-01	.202832-01
FUEL PREBURNER BYPASS CIRCUIT								
WEIGHT FLOW RATE	LB/SEC							
INLET PRESSURE	PSIA							
INLET TEMPERATURE	R							
INLET DENSITY	LB/FT3							
PRESSURE DROP	PSI							
ADMITTANCE	KW							
FUEL COOL. JKT. BYPASS CIRCUIT								
WEIGHT FLOW RATE	LB/SEC							
INLET PRESSURE	PSIA	4417.02	3344.91	2400.19	1963.17	1550.32	1152.27	768.403
INLET TEMPERATURE	R	121.592	101.600	85.0914	78.1350	71.5490	65.0826	57.8414
INLET DENSITY	LB/FT3	4.06291	4.07370	4.05521	4.05194	4.01424	3.98632	3.99077
PRESSURE DROP	PSI	2423.14	1769.79	1231.61	994.243	778.945	576.606	386.677
ADMITTANCE	KW							
FUEL SHUTOFF VALVE								
WEIGHT FLOW RATE	LB/SEC	7.45720	5.99257	4.51280	3.76170	3.01966	2.27337	1.53150
INLET PRESSURE	PSIA	4475.79	3382.82	2421.82	1978.22	1560.11	1157.87	770.941
INLET TEMPERATURE	R	121.592	101.600	85.0914	78.1350	71.5490	65.0826	57.8414
INLET DENSITY	LB/FT3	4.08897	4.09335	4.06881	4.06365	4.02144	3.99172	3.99343
PRESSURE DROP	PSI	58.7699	37.9110	21.6293	15.0476	9.79835	5.59496	2.53807
ADMITTANCE	KW	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:16
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:16
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71	14:18
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71	14:18
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19

TABLE LV (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7
ENGINE THRUST	LBS	25002.4	20046.5	15029.8	12493.3	9988.65	7492.36	4998.85
ENGINE MIXTURE RATIO	-	6.00014	6.00016	5.99954	5.99965	5.99985	6.00122	6.00158
SUMMARY FUEL PRESSURE SCHEDULE								
TANK OUTLET PRESSURE	PSIA	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100
TANK OUTLET TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
TANK OUTLET ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
BOOST PUMP SUCTION	PSIA	19.3069	20.0197	20.5786	20.8015	20.9836	21.1246	21.2262
DISCHARGE	PSIA	47.9020	42.6825	38.6714	36.8921	34.8350	32.6343	29.7569
HIGH SPEED INDUCER SUCTION	PSIA	47.9020	42.6825	38.6714	36.8921	34.8350	32.6343	29.7569
DISCHARGE	PSIA	198.118	160.717	129.250	113.695	97.4761	81.5105	63.2324
BOOST TURBINE INLET	PSIA	198.118	160.717	129.250	113.695	97.4761	81.5105	63.2324
DISCHARGE	PSIA	159.350	129.518	103.673	90.3149	76.4565	62.6928	47.4875
1ST STAGE PUMP SUCTION	PSIA	159.350	129.518	103.673	90.3149	76.4565	62.6928	47.4875
DISCHARGE	PSIA	1626.14	1225.57	892.302	735.468	580.811	441.115	288.106
2ND STAGE PUMP DISCHARGE	PSIA	3075.50	2310.41	1661.44	1361.19	1073.86	802.064	532.741
3RD STAGE PUMP DISCHARGE	PSIA	4475.79	3302.82	2421.82	1978.22	1560.11	1157.87	770.941
COOLING JACKET INLET	PSIA	4417.02	3344.91	2400.19	1963.17	1550.32	1152.27	768.403
DISCHARGE	PSIA	3558.08	2651.07	1868.75	1514.18	1181.06	866.153	565.189
PREBURNER INJECTOR INLET	PSIA	3538.27	2634.73	1855.95	1503.19	1171.94	858.920	559.949
SUMMARY OXIDIZER PRESSURE SCHEDULE								
TANK OUTLET PRESSURE	PSIA	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500
TANK OUTLET TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902
TANK OUTLET ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
BOOST PUMP SUCTION	PSIA	44.8915	45.5884	46.1349	46.3528	46.5309	46.6687	46.7681
DISCHARGE	PSIA	166.770	143.759	124.084	114.750	105.377	95.4325	82.8042
HIGH SPEED INDUCER SUCTION	PSIA	166.770	143.759	124.084	114.750	105.377	95.4325	82.8042
DISCHARGE	PSIA	785.917	617.631	480.490	415.843	352.958	289.717	217.130
BOOST TURBINE INLET	PSIA	785.917	617.631	480.490	415.843	352.958	289.717	217.130
DISCHARGE	PSIA	601.752	467.393	357.891	305.241	253.654	201.070	143.118
1ST STAGE PUMP SUCTION	PSIA	601.752	467.393	357.891	305.241	253.654	201.070	143.118
DISCHARGE	PSIA	[illegible]	2349.44	1695.22	1396.77	1107.03	836.022	559.526
THRUST CHAMBER VALVE INLET	PSIA	[illegible]	2349.44	1695.22	1396.77	1107.03	836.022	559.526
OX TC INJECTOR INLET	PSIA	[illegible]	1887.29	1327.63	1086.84	855.457	629.539	448.802
2ND STAGE PUMP SUCTION	PSIA	[illegible]	2233.42	1652.01	1373.01	1095.22	831.236	558.280
2ND STAGE PUMP DISCHARGE	PSIA	[illegible]	3622.07	2630.95	2162.99	1704.73	1277.77	842.136
PREBURNER VALVE INLET	PSIA	4665.91	3622.07	2630.95	2162.99	1704.73	1277.77	842.136
DISCHARGE	PSIA	3847.98	2683.28	1795.25	1411.56	1085.61	785.811	494.586
INJECTOR INLET	PSIA	3847.98	2683.28	1795.25	1411.56	1085.61	785.811	494.586
SUMMARY HOT GAS PRESSURE SCHEDULE								
PREBURNER INJECTOR FACE	PSIA	3271.25	2412.04	1678.76	1349.95	1042.86	755.399	483.170
TURBINE INLET		3215.41	2375.15	1656.30	1333.23	1031.00	747.597	478.805
TURBINE EXIT / INJECTOR INLET	PSIA	1993.88	1575.13	1168.58	968.926	771.371	575.664	381.726
TC INJECTOR FACE	PSIA	1654.00	1483.20	1112.40	927.000	741.600	556.200	370.800

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	15 JUL 71 14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71 14:16
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71 14:16
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	15 JUL 71 14:18
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	15 JUL 71 14:18
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19

TABLE LVI

PERFORMANCE DATA, 5:1 THROTTLING MIXTURE RATIO AND THRUST EXCURSIONS

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

SUMMARY ENGINE PERFORMANCE DATA

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
ENGINE SPECIFIC IMPULSE	SEC	462.466	465.669	467.018	457.811	464.231	449.812	455.215	459.372
TOTAL OXIDIZER FLOW RATE	LB/SEC	46.9206	46.0212	45.1885	23.6622	22.7450	9.62993	9.41290	9.20865
TOTAL FUEL FLOW RATE	LB/SEC	7.21769	7.67003	8.21718	3.64088	4.13499	1.48124	1.56840	1.67367
TOTAL FLOW LEAVING SYSTEM	LB/SEC	54.1335	53.6913	53.4057	27.3031	26.8800	11.1112	10.9813	10.8823
OX TANK OUTFLOW	LB/SEC	46.9195	46.0209	45.1909	23.6627	22.7443	9.63069	9.41285	9.20853
FUEL TANK OUTFLOW	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
OXIDIZER PRESSURE-INTERFACE	PSIA	46.3390	46.3584	46.3760	46.7200	46.7299	46.8285	46.8294	46.8303
OXIDIZER ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
OXIDIZER TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
OXIDIZER DENSITY	LB/FT3	71.0885	71.0884	71.0884	71.0877	71.0877	71.0874	71.0874	71.0874
FUEL PRESSURE-INTERFACE	PSIA	20.6668	20.8096	20.7357	21.1973	21.1645	21.2913	21.2891	21.2862
FUEL ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
FUEL TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
FUEL DENSITY	LB/FT3	4.34944	4.34937	4.34929	4.34980	4.34977	4.34991	4.34990	4.34990
OXID. BOOST PUMP SPEED	RPM	13703.4	14462.4	15459.5	9220.81	10152.1	6097.33	6308.89	6547.25
OXID. MAIN PUMP SPEED	RPM	47429.6	49869.7	52577.0	30381.6	32771.3	18476.0	18969.0	19522.7
FUEL BOOST PUMP SPEED	RPM	22469.7	23150.6	24469.9	15299.8	15686.2	10128.6	10178.3	10239.5
FUEL MAIN PUMP SPEED	RPM	77266.5	80058.2	83331.3	49701.3	51854.8	30247.1	30644.6	31108.3
OXID PUMP DISCH PRESS-STG1	PSIA	2684.87	3105.69	3599.86	1290.11	1529.33	532.174	559.526	590.277
OXID PUMP DISCH PRESS-STG2	PSIA	4101.00	4665.91	5302.16	2005.13	2360.93	800.506	842.136	889.355
FUEL PUMP DISCH PRESS STG3	PSIA	4257.70	4475.79	4756.37	1910.52	2061.51	749.146	770.941	796.148
PRIMARY COMBUSTOR PRESSURE	PSIA	3121.27	3271.25	3472.97	1313.91	1396.50	474.295	483.170	494.025
THRUST CHAMBER PRESSURE	PSIA	1784.34	1800.00	1818.25	892.170	909.126	356.868	360.000	363.650
OX. TC VALVE ADMITTANCE	KW	2.33486	1.48840	1.15847	1.45917	.943133	.824600	.804648	.688703
OX. PB VALVE ADMITTANCE	KW	.250641	.198186	.178679	.652661-01	.588563-01	.211711-01	.202832-01	.196320-01
FU. CJ BYP. VALVE ADMITTANCE	KW								
FU. PB BYP. VALVE ADMITTANCE	KW								

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
THRUST CHAMBER PERFORMANCE DATA									
THRUST CHAMBER PRESSURE	PSIA	1784.34	1800.00	1818.25	892.170	909.126	356.868	360.000	363.650
TC INJECTOR MIXTURE RATIO	-	6.60768	6.09389	5.58022	6.60514	5.58105	6.60560	6.09326	5.58157
TC AREA RATIO	-	270.000	270.000	270.000	270.000	270.000	270.000	270.000	270.000
TC SPECIFIC IMPULSE	SEC	463.042	466.263	467.624	458.373	464.827	450.347	455.773	459.950
TC THRUST	LBS	25014.2	24979.2	24918.1	12488.2	12466.8	4993.35	4994.22	4994.36
DUMP COOLANT THRUST	LBS	22.9266	23.1278	23.3623	11.4633	11.6812	4.58532	4.62556	4.67246
DUMP COOLANT FLOW RATE	LB/SEC	.116972	.117999	.119196	.584862-01	.595978-01	.233945-01	.235998-01	.238391-01
REGEN COOLANT FLOW RATE	LB/SEC	6.89817	7.33920	7.87145	3.48241	3.96956	1.42226	1.50790	1.61157
INLET PRESSURE	PSIA	4205.73	4417.02	4688.78	1897.10	2044.30	746.872	768.403	793.270
INLET TEMPERATURE	R	117.053	121.592	127.876	77.6086	79.1359	57.9114	57.8414	57.7741
INLET DENSITY	LB/FT3	4.06950	4.06291	4.05210	4.02615	4.06722	3.97060	3.99077	4.01356
INLET ENTHALPY	BTU/LB	220.600	238.928	264.428	50.2967	57.4540	-32.4815	-32.3148	-32.0810
PRESSURE DROP	PSI	812.829	858.942	911.676	427.614	473.605	194.732	203.214	212.829
ENTHALPY RISE	BTU/LB	1350.63	1279.33	1197.78	1538.55	1368.51	1828.83	1740.61	1637.82
TEMPERATURE RISE	R	344.246	324.935	303.475	396.478	351.325	477.004	452.730	424.453
OUTLET PRESSURE	PSIA	3392.90	3558.08	3777.10	1469.49	1570.70	552.140	565.189	580.441
OUTLET TEMPERATURE	R	461.299	446.527	431.352	474.086	430.461	534.915	510.572	482.227
OUTLET DENSITY	LB/FT3	1.18970	1.27556	1.38271	.545512	.636053	.189662	.203504	.221098
OUTLET ENTHALPY	BTU/LB	1571.23	1518.25	1462.20	1588.84	1425.96	1796.35	1708.30	1605.74
GAS INJECTOR FLOW RATE	LB/SEC	12.7713	13.5627	14.5676	5.33125	5.97310	1.86435	1.95644	2.07043
INLET PRESSURE	PSIA	1964.01	1993.88	2031.38	957.883	982.234	377.881	381.726	386.260
PRESSURE DROP	PSI	126.144	139.885	158.584	38.9474	45.8343	10.3068	10.9258	11.7001
ΔP/PCFACE	-	.686362-01	.754503-01	.846776-01	.423832-01	.489473-01	.280400-01	.294654-01	.312370-01
TC OXID. INJECTOR FLOW RATE	LB/SEC	41.1468	39.9087	38.6191	21.8661	20.7969	9.20600	8.98257	8.76846
INLET PRESSURE	PSIA	2406.65	2466.69	2612.61	1090.47	1098.15	421.500	448.802	446.180
INLET TEMPERATURE	R	181.712	184.158	187.765	171.637	174.575	167.232	167.728	168.310
INLET ENTHALPY	BTU/LB	-45.4046	-43.7952	-41.5902	-51.3592	-50.0108	-54.3061	-54.0693	-53.7933
OUTLET TEMPERATURE	R	524.409	552.825	591.605	346.990	356.557	257.305	260.257	258.915
OUTLET DENSITY	LB/FT3	11.9906	11.1788	10.0657	11.3488	10.9697	6.44105	6.62923	6.79487
PRESSURE DROP	PSI	568.778	612.692	739.808	171.536	161.753	53.9259	78.0025	71.6202
ΔP/PCFACE	-	.309477	.330470	.395028	.186669	.172739	.146708	.210363	.191212
TC IGNITER OXID. FLOW RATE	LB/SEC	.565463-01	.688548-01	.801888-01	.261659-01	.342604-01	.111858-01	.121053-01	.131175-01
FUEL FLOW RATE	LB/SEC	.996002-01	.101262	.103549	.479814-01	.493424-01	.179769-01	.184519-01	.190063-01
MIXTURE RATIO	-	.567733	.679965	.774400	.545334	.694341	.622233	.656045	.690164

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
PREBURNER PERFORMANCE DATA									
PREBURNER PRESSURE	PSIA	3121.27	3271.25	3472.97	1313.91	1396.50	474.295	483.170	494.025
2 MIXTURE RATIO	-	.816123	.811086	.812044	.500932	.475164	.286948	.273871	.261560
? GAS TEMPERATURE	R	1900.84	1878.82	1866.02	1380.33	1285.52	1046.03	998.154	947.636
SPECIFIC HT. RATIO	-	1.36396	1.36418	1.36406	1.38001	1.38121	1.38968	1.39021	1.39070
GAS CONSTANT	FT	421.293	422.473	422.248	510.674	519.703	596.862	603.088	609.072
FLOW RATE	LB/SEC	12.7161	13.4942	14.4851	5.30421	5.93959	1.85356	1.94489	2.05815
PB OXID. INJECTOR FLOW RATE	LB/SEC	5.64372	5.95248	6.37857	1.72646	1.86000	.395141	.398419	.405810
INLET PRESSURE	PSIA	3641.72	3847.98	4144.53	1375.24	1457.81	488.008	494.586	503.556
INLET TEMPERATURE	R	201.829	206.445	212.954	184.790	189.268	176.468	177.520	178.600
INLET DENSITY	LB/FT3	68.1440	67.4570	66.6730	68.3059	67.4488	68.0109	68.7369	69.5988
TEMPERATURE RISE	R	202.520	205.124	210.273	142.811	138.364	161.639	144.638	125.906
ENTHALPY RISE	BTU/LB	84.3271	83.7229	83.7462	81.0535	75.7159	113.771	108.534	102.617
OUTLET TEMPERATURE	R	404.349	411.569	423.227	327.601	327.631	338.108	322.157	304.506
OUTLET DENSITY	LB/FT3	30.6096	30.6872	30.5757	21.8444	24.4124	5.13074	5.66131	6.30049
PRESSURE DROP	PSI	520.456	576.726	671.564	61.3326	61.3018	13.7127	11.4163	9.53049
ΔP/PCPB	-	.166745	.176301	.193369	.466794-01	.438967-01	.289117-01	.236280-01	.192915-01
PB FUEL INJECTOR FLOW RATE	LB/SEC	6.89711	7.33825	7.87126	3.48191	3.96974	1.42216	1.50872	1.61087
INLET PRESSURE	PSIA	3374.15	3538.27	3756.08	1459.06	1559.08	547.138	559.949	574.932
INLET TEMPERATURE	R	461.299	446.527	431.352	474.086	430.461	534.915	510.572	482.227
INLET DENSITY	LB/FT3	1.18381	1.26919	1.37620	.541872	.631634	.187953	.201627	.219012
INLET ENTHALPY	BTU/LB	1571.23	1518.25	1462.20	1588.84	1425.96	1796.35	1708.30	1605.74
PRESSURE DROP	PSI	252.876	267.024	283.118	145.150	162.578	72.8434	76.7790	80.9071
ΔP/PCPB	-	.810170-01	.816275-01	.815204-01	.110472	.116418	.153583	.158907	.163771
PB IGNITER OXID. FLOW RATE	LB/SEC	.708801-01	.908130-01	.112754	.438286-01	.533372-01	.186010-01	.196862-01	.208978-01
FUEL FLOW RATE	LB/SEC	.103547	.111696	.122307	.514355-01	.568568-01	.180045-01	.188582-01	.198666-01
MIXTURE RATIO	-	.684520	.813034	.921891	.852109	.938097	1.03313	1.04391	1.05191

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
FUEL BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	19.5359	19.3069	19.0113	20.8588	20.7278	21.2353	21.2262	21.2146
ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
TEMPERATURE	R	37.3083	37.1297	36.8992	37.7929	37.7941	37.7897	37.7898	37.7899
SHAFT SPEED	RPM	22469.7	23150.6	24469.9	15299.8	15686.2	10128.6	10178.3	10239.5
WEIGHT FLOW RATE	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
VOLUME FLOW RATE	GPM	745.088	791.766	890.091	375.640	426.723	152.870	161.864	172.746
HEAD RISE (ACTUAL)	FT	879.600	906.165	970.323	516.806	508.500	273.755	272.403	270.679
PRESSURE RISE	PSI	27.7497	28.5951	23.6449	16.2348	15.9712	8.57325	8.53070	8.47647
(Q/N)/(QD/ND)	-	.957108	.987155	1.04991	.708660	.785198	.435635	.459016	.486948
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3797.41	3944.17	4195.57	2735.74	3026.03	1860.77	1931.25	2016.70
EFFICIENCY	-	.704591	.709794	.732642	.653131	.673075	.506437	.521732	.539198
NPSH (NO TSH)	FT	48.5949	40.7396	24.3073	94.1313	89.6954	106.874	106.567	106.173
NPSP (NO TSH)	PSI	1.46728	1.23003	.699327	2.84318	2.70910	3.22836	3.21909	3.20718
THERMODYNAMIC SUCT. HEAD	FT	122.345	120.569	118.278	127.162	127.173	127.130	127.130	127.131
SUCTION SPECIFIC SPEED		12973.8	14391.9	17692.7	5168.26	5733.81	2093.13	2166.50	2254.44
S/SDESIGN	-	.288308	.319819	.393171	.114850	.127418	.465141-01	.481444-01	.500988-01
S/SMAX AT Q/QD	-	.290736	.323109	.496802	.136247	.138161	.100772	.102083	.100187
SHAFT HORSEPOWER	HP	16.3842	17.8040	19.7856	5.23739	5.68035	1.45619	1.48926	1.52817
DISCHARGE PRESSURE	PSIA	27.7497	28.5951	23.6449	16.2348	15.9712	8.57325	8.53070	8.47647
ENTHALPY	BTU/LB	-104.656	-104.619	-104.558	-105.243	-105.289	-105.565	-105.589	-105.615
TEMPERATURE	R	38.0966	38.1035	38.1955	37.9899	37.9765	37.9551	37.9462	37.9367
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
INLET PRESSURE	PSIA	191.056	198.118	199.606	113.018	114.683	62.9263	63.2324	63.5522
ENTHALPY	BTU/LB	-96.3780	-96.0091	-95.5630	-100.771	-100.800	-103.142	-103.197	-103.257
TEMPERATURE	R	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337
PRESSURE DROP	PSI	37.6809	38.7681	40.1662	23.9371	22.8678	16.3027	15.7449	15.1361
ISEN. SPOUTING VELOCITY	FT/SEC	324.173	328.092	334.961	260.160	252.932	226.624	221.045	214.983
MEAN WHEEL SPEED	FT/SEC	199.674	205.725	217.449	135.959	139.393	90.0068	90.4477	90.9916
VELOCITY RATIO - U/CO	-	.615948	.627031	.649174	.522596	.551109	.397163	.409181	.423249
EFFICIENCY	-	.764460	.763447	.759862	.752268	.759921	.677572	.687625	.698613
SHAFT HORSEPOWER	HP	16.3851	17.8106	19.7931	5.23731	5.68052	1.45684	1.48931	1.52748
OUTLET PRESSURE	PSIA	153.177	159.350	159.440	89.0805	91.8148	46.6235	47.4875	48.4161
ENTHALPY	BTU/LB	-97.9824	-97.6503	-97.2656	-101.788	-101.771	-103.837	-103.868	-103.902
TEMPERATURE	R	39.6719	39.7519	39.9205	38.7651	38.7396	38.4378	38.4145	38.3892
TURBINE TORQUE	FT-LBS	3.82969	4.04063	4.24830	1.79787	1.90197	.755431	.768503	.783489

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
FUEL PUMP - HIGH SPEED INDUCER AND FIRST STAGE									
HIGH SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	47.2856	47.9020	42.6562	37.0936	36.6990	29.8085	29.7569	29.6910
ENTHALPY	BTU/LB	-104.656	-104.619	-104.558	-105.243	-105.289	-105.565	-105.589	-105.615
TEMPERATURE	R	38.0868	38.0924	38.1828	37.9876	37.9735	37.9547	37.9458	37.9362
SHAFT SPEED	RPM	77266.5	80058.2	83331.3	49701.3	51854.8	30247.1	30644.6	31108.3
WEIGHT FLOW RATE	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
VOLUME FLOW RATE	GPM	744.702	791.302	849.032	375.653	426.687	152.974	161.959	172.832
HEAD RISE (ACTUAL)	FT	4554.32	4781.13	5031.05	2317.02	2379.39	1035.44	1046.74	1058.95
PRESSURE RISE	PSI	143.772	150.216	156.950	75.9241	77.9836	33.1178	33.4755	33.8611
(Q/N)/(QD/ND)	-	.973545	.998392	1.02916	.763457	.831160	.510856	.533847	.561194
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3803.34	3916.78	4041.07	2884.45	3144.09	2049.51	2119.22	2203.10
EFFICIENCY	-	.707016	.713557	.718739	.665780	.681135	.549174	.562396	.577101
NPSH (NO TSH)	FT	942.031	961.897	780.958	614.018	602.199	376.044	375.117	373.776
NPSP (NO TSH)	PSI	28.4585	29.0590	23.5549	18.5453	18.1900	11.3516	11.3245	11.2851
THERMODYNAMIC SUCT. HEAD	FT	130.063	130.139	131.037	129.096	128.956	128.770	128.681	128.586
SUCTION SPECIFIC SPEED		11253.9	11854.9	14631.1	6768.14	7617.93	3512.73	3667.46	3854.13
S/SDESIGN	-	.375129	.395165	.487702	.225605	.253931	.117091	.122249	.128471
S/SMAX AT Q/QD	-	.378492	.399893	.561208	.249877	.265880	.219573	.214598	.208650
SHAFT HORSEPOWER	HP	84.5418	93.4423	104.571	23.0349	26.2652	5.07920	5.30891	5.58583
DISCHARGE PRESSURE	PSIA	191.058	198.118	199.606	113.018	114.683	62.9263	63.2324	63.5522
ENTHALPY	BTU/LB	-96.3780	-96.0091	-95.5630	-100.771	-100.800	-103.142	-103.197	-103.257
TEMPERATURE	R	39.9871	40.0816	40.2687	38.9138	38.8820	38.5193	38.4930	38.4645
FUEL PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
VOLUME FLOW RATE (1)	GPM	865.060	921.564	992.634	427.848	482.477	178.373	187.762	199.110
HEAD RISE (ACTUAL)	FT	46893.9	49605.1	52898.4	20883.5	22474.7	7954.19	8183.15	8445.40
PRESSURE RISE	PSI	1385.96	1466.79	1565.82	622.758	673.240	233.644	240.619	248.629
(Q/N)/(QD/ND)		.978867	1.00644	1.04148	.752646	.813497	.515601	.535700	.559609
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	713.144	731.179	752.697	591.779	620.521	476.957	486.735	498.262
EFFICIENCY	-	.605561	.606118	.601119	.561794	.578964	.457887	.469699	.482842
NPSH (NO TSH)	FT	3416.16	3559.15	3464.54	1790.10	1883.44	642.100	677.984	716.593
NPSP (NO TSH)	PSI	95.2145	98.7807	95.4521	51.8134	54.5397	18.8966	19.9658	21.1180
THERMODYNAMIC SUCT. HEAD	FT	215.323	219.274	224.266	177.089	177.107	155.603	155.103	154.556
SUCTION SPECIFIC SPEED		4857.93	5042.96	5546.75	3480.39	3724.26	2691.33	2707.95	2737.50
S/SDESIGN	-	.485793	.504296	.554675	.348039	.372426	.269133	.270795	.273750
S/SMAX AT Q/QD	-	.476591	.507998	.587869	.404849	.401645	.469539	.462467	.450685
SHAFT HORSEPOWER	HP	1089.22	1221.14	1403.97	268.551	316.393	53.1979	56.1983	59.8735
DISCHARGE PRESSURE	PSIA	1539.14	1626.14	1725.26	711.839	765.055	280.268	288.106	297.045
ENTHALPY	BTU/LB	21.3181	28.2682	37.7028	-43.7650	-41.5648	-76.0619	-76.0970	-76.1186
TEMPERATURE	R	70.7292	72.3663	74.7842	54.5136	54.8595	46.8368	46.7897	46.7249

(1) VOLUME FLOW INCLUDES 7% RECIRCULATION

CASE 1 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
FUEL PUMP - SECOND AND THIRD STAGES									
SECOND STAGE FUEL PUMP									
SUCTION PRESSURE	PSIA	1539.14	1626.14	1725.26	711.839	765.055	280.268	288.106	297.045
ENTHALPY	BTU/LB	21.3181	28.2682	37.7028	-43.7650	-41.5648	-76.0619	-76.0970	-76.1186
TEMPERATURE	R	70.7100	72.3447	74.7559	54.5093	54.8534	46.8357	46.7884	46.7240
SHAFT SPEED	RPM	77266.5	80058.2	83331.3	49701.3	51854.8	30247.1	30644.6	31108.3
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
VOLUME FLOW RATE (1)	GPM	859.465	913.890	981.236	435.464	488.744	182.997	192.290	203.512
HEAD RISE (ACTUAL)	FT	47028.7	49760.6	53174.0	20828.9	22417.7	8005.03	8203.23	8434.86
PRESSURE RISE	PSI	1362.67	1449.36	1542.38	604.329	652.663	237.910	244.635	252.416
(Q/N)/(QD/ND)	-	.973600	.999153	1.03065	.766881	.824965	.529546	.549219	.572607
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	709.305	726.422	745.453	598.195	625.730	483.486	492.996	504.212
EFFICIENCY	-	.605497	.607008	.602704	.565987	.580134	.468039	.477641	.488532
SHAFT HORSEPOWER	HP	1092.47	1223.18	1407.58	265.865	314.953	52.3767	55.3995	59.1022
DISCHARGE PRESSURE	PSIA	2901.81	3075.50	3267.65	1316.17	1417.72	518.178	532.741	549.461
ENTHALPY	BTU/LB	121.127	133.612	151.077	3.52594	8.09218	-54.0834	-54.0271	-53.9314
TEMPERATURE	R	94.2464	97.2033	101.655	66.2552	67.1787	52.5215	52.4831	52.4505
THIRD STAGE FUEL PUMP									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	7.67015	8.21652	3.64041	4.13533	1.48165	1.56881	1.67428
VOLUME FLOW RATE (1)	GPM	856.200	905.296	972.386	440.001	492.371	186.710	195.950	207.099
HEAD RISE (ACTUAL)	FT	46876.3	49747.2	53152.6	20768.8	22338.3	7994.29	8191.14	8421.10
PRESSURE RISE	PSI	1355.89	1400.29	1488.72	594.358	643.791	230.967	238.200	246.687
(Q/N)/(QD/ND)	-	.979547	.999600	1.03151	.782577	.839354	.545663	.565241	.588494
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	709.682	723.144	742.308	602.610	629.720	483.857	498.218	509.259
EFFICIENCY	-	.605567	.607005	.602579	.570630	.581526	.475559	.484793	.495253
SHAFT HORSEPOWER	HP	1086.60	1222.85	1407.30	262.940	313.086	51.4793	54.5016	58.2051
FUEL TURBINE SPEED	RPM	77266.5	80058.2	83331.3	49701.3	51854.8	30247.1	30644.6	31108.3
COMBINED FUEL PUMP PERFORMANCE BASED ON DELIVERED FLOW									
TOTAL FULL TPA POWER REQ'D	HP	3355.03	3760.62	4323.42	820.392	970.697	162.133	171.408	182.767
TOTAL FULL TPA HEAD RISE	FT	145353.	153894.	164256.	64798.3	69610.2	24988.9	25624.3	26360.3
OVERALL PUMP EFFICIENCY	-	.568598	.570695	.567570	.522793	.539184	.415201	.426409	.439054

(1) VOL. FLOW INCLUDES 7% RECIRC.

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   5:1 THROTTLEABLE   MIXTURE RATIO AND THRUST EXCURSIONS   PAGE 7

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
OXIDIZER BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	44.6143	44.8915	44.9615	46.3322	46.3716	46.7642	46.7681	46.7716
ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
TEMPERATURE	R	162.928	162.927	162.927	162.915	162.914	162.904	162.904	162.904
SHAFT SPEED	RPM	13703.4	14462.4	15459.5	9220.81	10152.1	6097.33	6308.89	6547.25
WEIGHT FLOW RATE	LB/SEC	46.9195	46.0209	45.1909	23.6627	22.7443	9.63069	9.41285	9.20853
VOLUME FLOW RATE	GPM	296.201	290.529	285.290	149.389	143.591	60.8018	59.4265	58.1366
HEAD RISE (ACTUAL)	FT	208.407	246.882	309.205	121.578	160.802	66.9665	72.9974	79.9535
PRESSURE RISE	PSI	102.885	121.878	152.645	60.0187	79.3822	33.0589	36.0361	39.4701
(Q/N)/(QD/ND)	-	1.00510	.989884	.909338	.798331	.696955	.491373	.464153	.437547
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4299.70	3957.92	3541.23	3078.13	2694.03	2030.98	1947.43	1867.05
EFFICIENCY	-	.671241	.660592	.655669	.624099	.604141	.498826	.484478	.469745
NPSH (NO TSH)	FT	60.0662	60.2236	60.3663	63.1627	63.2441	64.0549	64.0628	64.0700
NPSP (NO TSH)	PSI	29.6542	29.7319	29.8023	31.1815	31.2217	31.6216	31.6255	31.6291
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		10414.8	10865.9	11490.7	4803.76	5180.60	2006.57	2052.39	2106.52
S/SDESIGN	-	.231440	.241464	.255348	.106750	.115124	.445904-01	.456088-01	.468117-01
S/SMAX AT Q/QD	-	.313594	.244034	.258840	.114448	.139016	.881735-01	.958714-01	.101373
SHAFT HORSEPOWER	HP	26.4865	31.2715	38.7481	8.38114	11.0069	2.35074	2.57865	2.84973
DISCHARGE PRESSURE	PSIA	102.885	121.878	152.645	60.0187	79.3822	33.0589	36.0361	39.4701
ENTHALPY	BTU/LB	-56.6460	-56.5647	-56.4390	-56.7947	-56.7030	-56.8725	-56.8514	-56.8263
TEMPERATURE	R	163.046	163.119	163.222	163.051	163.083	163.004	163.017	163.035
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	46.9195	46.0209	45.1909	23.6627	22.7443	9.63069	9.41285	9.20853
INLET PRESSURE	PSIA	666.861	785.917	933.671	377.122	466.362	204.956	217.130	231.037
ENTHALPY	BTU/LB	-54.5909	-54.1338	-53.5026	-55.6750	-55.2608	-56.2603	-56.1791	-56.0835
TEMPERATURE	R	165.309	165.309	165.309	165.309	165.309	165.309	165.309	165.309
PRESSURE DROP	PSI	152.485	184.164	232.160	95.6536	130.810	65.9479	74.0127	83.6084
ISEN. SPOUTING VELOCITY	FT/SEC	161.777	177.347	199.355	128.874	151.977	112.582	120.263	128.958
MEAN WHEEL SPEED	FT/SEC	101.606	107.233	114.626	68.3687	75.2740	45.2094	46.7780	48.5453
VELOCITY RATIO - U/CO	-	.628057	.604651	.574984	.530505	.495296	.401567	.388963	.376442
EFFICIENCY	-	.763327	.764954	.763670	.754737	.741704	.681327	.670360	.658797
SHAFT HORSEPOWER	HP	26.4851	31.2852	38.7537	8.38099	11.0094	2.34992	2.57867	2.85062
OUTLET PRESSURE	PSIA	514.376	631.752	701.511	281.469	335.551	139.008	143.118	147.429
ENTHALPY	BTU/LB	-54.9899	-54.6143	-54.1087	-55.9253	-55.6029	-56.4327	-56.3727	-56.3023
TEMPERATURE	R	165.591	166.225	167.140	164.325	164.813	163.646	163.749	163.877
TURBINE TORQUE	FT-LBS	10.1509	11.3614	13.1659	4.77376	5.69563	2.02418	2.14673	2.28673

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71 14:17
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:17
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 14:19
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 14:20

TABLE LVI (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
OXIDIZER PUMP - HIGH SPEED INDUCER AND MAIN STAGE									
HIGH SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	147.699	166.770	197.607	106.351	125.754	79.8231	82.8042	86.2417
ENTHALPY	BTU/LB	-56.0460	-56.5647	-56.4390	-56.7947	-56.7030	-56.8725	-56.8514	-56.8263
TEMPERATURE	R	163.070	163.120	163.223	163.065	163.079	163.006	163.019	163.037
SHAFT SPEED	RPM	47429.6	49869.7	52577.0	30381.6	32771.3	18476.0	18969.0	19522.7
WEIGHT FLOW RATE	LB/SEC	46.9195	46.0209	45.1909	23.6627	22.7443	9.63069	9.41285	9.20853
VOLUME FLOW RATE	GPM	297.030	291.432	286.310	149.619	143.943	60.8981	59.5322	58.2534
HEAD RISE (ACTUAL)	FT	1054.53	1258.01	1496.27	549.331	691.643	253.880	272.585	293.897
PRESSURE RISE	PSI	519.162	619.147	736.065	270.772	340.608	125.133	134.326	144.795
(Q/N)/(QD/ND)	-	1.07975	1.00756	.938885	.849076	.757301	.568285	.541102	.514461
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4400.60	4030.34	3697.90	3275.13	2915.26	2266.94	2181.70	2099.20
EFFICIENCY	-	.659387	.665005	.654806	.630473	.616281	.532885	.521027	.508478
NPSH (NO TSH)	FT	269.002	307.759	370.433	184.798	224.346	131.076	137.131	144.114
NPSP (NO TSH)	PSI	132.434	151.467	182.228	91.0891	110.482	64.6054	67.5765	71.0011
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		12171.0	11474.7	10451.6	7296.35	6693.36	3638.95	3574.40	3509.57
S/SDESIGN	-	.405699	.382490	.348387	.243212	.223112	.121298	.119147	.116986
S/SMAX AT Q/QD	-	.388662	.400133	.351593	.252055	.248753	.193193	.204827	.217143
SHAFT HORSEPOWER	HP	136.430	158.289	187.753	37.4860	46.4102	8.34237	8.95363	9.67723
DISCHARGE PRESSURE	PSIA	666.861	785.917	933.671	377.122	466.362	204.956	217.130	231.037
ENTHALPY	BTU/LB	-54.5909	-54.1338	-53.5026	-55.6750	-55.2608	-56.2603	-56.1791	-56.0835
TEMPERATURE	R	166.057	167.010	167.810	164.393	165.270	163.602	163.717	163.925
OXIDIZER PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	46.9195	46.0209	45.1909	23.6627	22.7443	9.63069	9.41285	9.20853
VOLUME FLOW RATE (1)	GPM	312.739	307.888	303.724	159.192	154.332	66.4797	65.2751	64.1776
HEAD RISE (ACTUAL)	FT	4424.90	5105.12	5912.59	2053.21	2433.55	799.600	847.239	901.482
PRESSURE RISE	PSI	2170.50	2503.94	2898.35	1008.64	1193.78	393.165	416.408	442.849
5 (Q/N)/(QD/ND)		1.06450	1.01544	.950124	.861801	.774564	.591802	.565978	.540678
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	1546.01	1448.87	1358.95	1256.74	1175.01	1001.84	975.919	950.637
EFFICIENCY	-	.620123	.635209	.636649	.606266	.588442	.510837	.500242	.489177
NPSH (NO TSH)	FT	1014.34	1192.19	1395.72	539.837	650.169	250.539	258.832	267.519
NPSP (NO TSH)	PSI	496.347	583.075	681.872	264.860	318.424	123.101	127.111	131.300
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		4652.66	4302.12	4004.09	3403.85	3147.42	2363.95	2347.80	2338.20
S/SDESIGN	-	.465266	.430212	.400409	.340385	.314742	.236395	.234780	.233820
S/SMAX AT Q/QD	-	.537138	.438299	.388476	.350323	.355466	.366570	.382239	.396616
SHAFT HORSEPOWER	HP	637.024	706.006	803.911	154.309	182.355	29.8273	31.6727	33.8635
DISCHARGE PRESSURE	PSIA	2684.87	3105.69	3599.86	1290.11	1529.33	532.174	559.526	590.277
ENTHALPY	BTU/LB	-45.4046	-43.7952	-41.5902	-51.3592	-50.0108	-54.3061	-54.0693	-53.7933
TEMPERATURE	R	181.712	184.158	187.765	171.637	174.575	167.232	167.728	168.310

(1) VOLUME FLOW INCLUDES 5% RECIRCULATION

CASE 1	LETS2 O.S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 O.S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 3	LETS2 O.S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 4	LETS2 O.S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71 14:17
CASE 5	LETS2 O.S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:17
CASE 6	LETS2 O.S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 14:19
CASE 7	LETS2 O.S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19
CASE 8	LETS2 O.S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
OXIDIZER PUMP - HALF STAGE									
SUCTION PRESSURE	PSIA	2452.84	2846.36	3300.23	1267.85	1503.32	530.958	558.280	588.979
ENTHALPY	BTU/LB	-45.4046	-43.7952	-41.5902	-51.3592	-50.0108	-54.3061	-54.0693	-53.7933
TEMPERATURE	R	183.799	186.539	190.474	172.588	175.786	168.018	168.566	169.205
SHAFT SPEED	RPM	47429.6	49869.7	52577.0	30381.6	32771.3	18476.0	18969.0	19522.7
WEIGHT FLOW RATE	LB/SEC	5.71434	6.04331	6.49130	1.77026	1.91320	.413233	.418135	.426717
VOLUME FLOW RATE	GPM	38.6798	40.8821	43.9716	12.5810	13.5971	3.41924	3.47391	3.55563
HEAD RISE (ACTUAL)	FT	3360.26	3713.59	4091.64	1505.46	1751.51	551.478	581.162	615.511
PRESSURE RISE	PSI	1648.16	1819.55	2001.93	737.285	857.611	269.549	283.855	300.376
(Q/N)/(QD/ND)	-	.926729	.931567	.950373	.470567	.471486	.210299	.208109	.206963
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	667.468	670.282	681.488	445.879	446.331	300.212	298.697	297.901
EFFICIENCY	-	.452608	.454885	.455532	.336195	.339026	.193188	.192281	.192086
SHAFT HORSEPOWER	HP	81.5064	94.5882	111.677	16.0455	20.0029	2.77793	2.98581	3.23679
DISCHARGE PRESSURE	PSIA	4101.00	4665.91	5302.16	2005.13	2360.93	800.506	842.136	889.355
ENTHALPY	BTU/LB	-35.3751	-32.7854	-29.4811	-45.2254	-42.9312	-50.3236	-49.8503	-49.3184
TEMPERATURE	R	201.829	206.445	212.954	184.790	189.268	176.468	177.520	178.600

CASE 1	LETS2 COS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 COS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 3	LETS2 COS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 4	LETS2 COS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71 14:17
CASE 5	LETS2 COS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:17
CASE 6	LETS2 COS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 14:19
CASE 7	LETS2 COS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19
CASE 8	LETS2 COS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 14:20

TABLE LVI (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 5:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
GAS TURBINE PERFORMANCE DATA									
FUEL TPA SHAFT SPEED	RPM	77266.5	80058.2	83331.3	[illegible]9701.3	51854.8	30247.1	30644.6	31108.3
TURBINE INLET TEMPERATURE	R	1882.40	1860.66	1847.88	1369.26	1275.72	1037.17	990.101	940.373
WEIGHT FLOW RATE (1)	LB/SEC	9.45335	10.0318	10.7693	3.94371	4.41569	1.37801	1.44595	1.52993
MEAN WHEEL SPEED	FT/SEC	1254.56	1299.89	1353.04	806.992	841.958	491.117	497.572	505.101
ISEN. SPOUTING VELOCITY	FT/SEC	4636.24	4763.78	4929.32	3622.97	3714.49	2905.52	2904.36	2904.69
STAGE VEL. RATIO U/C	-	.382684	.385895	.388183	.315005	.320557	.239043	.242281	.245919
INLET PRESSURE (TOTAL)	PSIA	3070.41	3215.41	3410.63	1298.32	1378.28	470.156	478.805	489.376
EXIT PRESSURE (STATIC)	PSIA	1964.01	1993.88	2031.38	957.883	982.234	377.881	381.726	386.260
PRESSURE RATIO	-	1.56334	1.61264	1.67897	1.35541	1.40321	1.24419	1.25432	1.26696
SHAFT HORSEPOWER	HP	3355.22	3760.87	4323.89	820.194	970.894	162.194	171.348	182.843
SHAFT TORQUE	FT-LBS	228.068	246.727	272.522	86.6728	98.3369	28.1634	29.3669	30.8700
EXIT TEMPERATURE	R	1756.75	1730.36	1708.39	1307.58	1211.50	1006.76	959.767	910.054
EFFICIENCY	-	.626218	.628508	.628664	.602819	.606289	.530447	.534479	.538908
OXID TPA SHAFT SPEED	RPM	47429.6	49869.7	52577.0	30381.6	32771.3	18476.0	18969.0	19522.7
TURBINE INLET TEMPERATURE	R	1882.40	1860.66	1847.88	1369.26	1275.72	1037.17	990.101	940.373
WEIGHT FLOW RATE (1)	LB/SEC	3.20236	3.46200	3.71651	1.36098	1.52386	.475552	.498999	.527980
MEAN WHEEL SPEED	FT/SEC	1042.62	1096.26	1155.77	667.863	720.395	406.149	416.986	429.158
ISEN. SPOUTING VELOCITY	FT/SEC	4636.24	4763.78	4929.32	3622.97	3714.49	2905.52	2904.36	2904.69
VELOCITY RATIO - U/C	-	.224864	.230123	.234468	.184341	.193941	.139785	.143572	.147746
INLET PRESSURE (TOTAL)	PSIA	3070.41	3215.41	3410.63	1298.32	1378.28	470.156	478.805	489.376
EXIT PRESSURE (STATIC)	PSIA	1964.01	1993.88	2031.38	957.883	982.234	377.881	381.726	386.260
PRESSURE RATIO	-	1.56334	1.61264	1.67897	1.35541	1.40321	1.24419	1.25432	1.26696
SHAFT HORSEPOWER	HP	854.774	959.195	1103.14	207.800	248.718	40.8995	43.5412	46.8173
SHAFT TORQUE	FT-LBS	94.6534	101.019	110.197	35.9227	39.8609	11.6263	12.0556	12.5950
EXIT TEMPERATURE	R	1791.12	1764.36	1744.70	1323.98	1228.05	1014.95	967.765	917.878
EFFICIENCY	-	.463760	.464497	.464761	.442557	.450058	.387595	.393556	.399848
TURBINE GAS PROPERTIES									
SPECIFIC HEAT RATIO	-	1.36396	1.36418	1.36406	1.38001	1.38121	1.38968	1.39021	1.39070
GAS CONSTANT	FT	421.293	422.473	422.248	510.674	519.703	596.862	603.088	609.072
MOLECULAR WEIGHT	-	3.66491	3.65467	3.65662	3.02345	2.97093	2.58686	2.56016	2.53501
TEMPERATURE	R	1882.40	1860.66	1847.88	1369.26	1275.72	1037.17	990.101	940.373

(1) INCLUDES 7% LEAKAGE FLOW

CASE 1 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 3 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 4 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:17
CASE 5 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:17
CASE 6 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	14:19
CASE 7 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71	14:19
CASE 8 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	14:20

TABLE LVI (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   5:1 THROTTLEABLE   MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
CONTROLS COMPONENTS									
OXID. THRUST CHAMBER VALVE									
WEIGHT FLOW RATE	LB/SEC	41.1486	39.9088	38.6194	21.8662	20.7969	9.20622	8.98261	8.76869
INLET PRESSURE	PSIA	2684.67	3105.69	3599.86	1290.11	1529.33	532.174	559.526	590.277
INLET TEMPERATURE	R	181.712	184.158	187.765	171.637	174.575	167.232	167.728	168.310
INLET DENSITY	LB/FT3	70.5042	70.5557	70.4553	70.5227	70.5081	70.3836	70.3335	70.2735
PRESSURE DROP	PSI	274.869	635.843	984.272	198.698	430.324	110.507	110.564	143.946
9 ADMITTANCE	KW	2.33486	1.48840	1.15847	1.45917	.943133	.82460[illegible]	.804648	.688703
OXID. PREBURNER VALVE									
WEIGHT FLOW RATE	LB/SEC	5.64346	5.95250	6.37854	1.72644	1.85986	.394682	.398449	.405819
INLET PRESSURE	PSIA	4101.00	4665.91	5302.16	2005.13	2360.93	800.506	842.136	889.355
INLET TEMPERATURE	R	201.829	206.445	212.954	184.790	189.268	176.468	177.520	178.600
INLET DENSITY	LB/FT3	68.8806	68.8206	68.6928	69.3178	68.9937	69.3976	69.2848	69.1126
PRESSURE DROP	PSI	459.277	817.932	1157.63	629.891	903.129	312.499	347.549	385.799
P ADMITTANCE	KW	.250641	.198186	.178679	.652661-01	.588563-01	.211711-01	.202832-01	.196320-01
FULL PREBURNER BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC								
INLET PRESSURE	PSIA								
INLET TEMPERATURE	R								
INLET DENSITY	LB/FT3								
PRESSURE DROP	PSI								
ADMITTANCE	KW								
FUEL COOL. JKT. BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC								
INLET PRESSURE	PSIA	4205.73	4417.02	4688.78	1897.10	2044.30	746.872	768.403	793.270
INLET TEMPERATURE	R	117.053	121.592	127.878	77.6086	79.1359	57.9114	57.8414	57.7741
INLET DENSITY	LB/FT3	4.06950	4.06291	4.05210	4.02615	4.06722	3.97060	3.99077	4.01356
PRESSURE DROP	PSI	2241.72	2423.14	2657.39	939.219	1062.07	368.992	386.677	407.010
ADMITTANCE	KW								
FUEL SHUTOFF VALVE									
WEIGHT FLOW RATE	LB/SEC	7.01524	7.45720	7.99067	3.54100	4.02913	1.44566	1.53150	1.63540
INLET PRESSURE	PSIA	4257.70	4475.79	4756.37	1910.52	2061.51	749.146	770.941	796.148
INLET TEMPERATURE	R	117.053	121.592	127.878	77.6086	79.1359	57.9114	57.8414	57.7741
INLET DENSITY	LB/FT3	4.09248	4.08897	4.08212	4.03654	4.07765	3.97297	3.99343	4.01658
PRESSURE DROP	PSI	51.9656	58.7699	67.5923	13.4233	17.2041	2.27319	2.53807	2.87748
ADMITTANCE	KW	3.80000	3.60000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000

CASE 1	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71 14:13
CASE 3	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 4	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71 14:17
CASE 5	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:17
CASE 6	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 14:19
CASE 7	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	15 JUL 71 14:19
CASE 8	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 14:20

TABLE LVI (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	25002.4	24941.4	12499.7	12478.5	4997.93	4998.85	4999.03
ENGINE MIXTURE RATIO	-	6.50059	6.00014	5.49928	6.49904	5.50061	6.50127	6.00158	5.50207
SUMMARY FUEL PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100
TANK OUTLET TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
TANK OUTLET ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
BOOST PUMP SUCTION	PSIA	19.5359	19.3069	19.0113	20.8588	20.7278	21.2353	21.2262	21.2146
DISCHARGE	PSIA	47.2856	47.9020	42.656	37.0936	36.6990	29.8085	29.7569	29.6910
HIGH SPEED INDUCER SUCTION	PSIA	47.2856	47.9020	42.6562	37.0936	36.6990	29.8085	29.7569	29.6910
DISCHARGE	PSIA	191.058	198.118	199.606	113.018	114.683	62.9263	63.2324	63.5522
BOOST TURBINE INLET	PSIA	191.058	198.118	199.606	113.018	114.683	62.9263	63.2324	63.5522
DISCHARGE	PSIA	153.177	159.350	159.440	89.0805	91.8148	46.6235	47.4875	48.4161
1ST STAGE PUMP SUCTION	PSIA	153.177	159.350	159.440	89.0805	91.8148	46.6235	47.4875	48.4161
DISCHARGE	PSIA	1539.14	1626.14	1725.26	711.839	765.055	280.268	288.106	297.045
2ND STAGE PUMP DISCHARGE	PSIA	2901.81	3075.50	3267.65	1316.17	1417.72	518.178	532.741	549.461
3RD STAGE PUMP DISCHARGE	PSIA	4257.70	4475.79	4756.37	1910.52	2061.51	749.146	770.941	796.148
COOLING JACKET INLET	PSIA	4205.73	4417.02	4688.78	1897.10	2044.30	746.872	768.403	793.270
DISCHARGE	PSIA	3392.90	3558.08	3777.10	1469.49	1570.70	552.140	565.189	580.441
PREBURNER INJECTOR INLET	PSIA	3374.15	3538.27	3756.08	1459.06	1559.08	547.138	559.949	574.932
SUMMARY OXIDIZER PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500
TANK OUTLET TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
TANK OUTLET ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
BOOST PUMP SUCTION	PSIA	44.8143	44.8915	44.9615	46.3322	46.3716	46.7642	46.7681	46.7716
DISCHARGE	PSIA	147.699	166.770	197.607	106.351	125.754	79.8231	82.8042	86.2417
HIGH SPEED INDUCER SUCTION	PSIA	147.699	166.770	197.607	106.351	125.754	79.8231	82.8042	86.2417
DISCHARGE	PSIA	666.861	785.917	933.671	377.122	466.362	204.956	217.130	231.037
BOOST TURBINE INLET	PSIA	666.861	785.917	933.671	377.122	466.362	204.956	217.130	231.037
DISCHARGE	PSIA	514.376	601.752	701.511	281.469	335.551	139.008	143.118	147.429
1ST STAGE PUMP SUCTION	PSIA	514.376	601.752	701.511	281.469	335.551	139.008	143.118	147.429
DISCHARGE	PSIA	2684.87	3105.69	3599.86	1290.11	1529.33	532.174	559.526	590.277
THRUST CHAMBER VALVE INLET	PSIA	2684.87	3105.69	3599.86	1290.11	1529.33	532.174	559.526	590.277
OX TC INJECTOR INLET	PSIA	2406.65	2466.69	2612.61	1090.47	1098.15	421.500	448.802	446.180
2ND STAGE PUMP SUCTION	PSIA	2452.84	2846.36	3300.23	1267.85	1503.32	530.958	558.280	588.979
2ND STAGE PUMP DISCHARGE	PSIA	4101.00	4665.91	5302.16	2005.13	2360.93	800.506	842.136	889.355
PREBURNER VALVE INLET	PSIA	4101.00	4665.91	5302.16	2005.13	2360.93	800.506	842.136	889.355
DISCHARGE	PSIA	3641.72	3847.98	4144.53	1375.24	1457.81	488.008	494.586	503.556
INJECTOR INLET	PSIA	3641.72	3847.98	4144.53	1375.24	1457.81	488.008	494.586	503.556
SUMMARY HOT GAS PRESSURE SCHEDULE									
PREBURNER INJECTOR FACE	PSIA	3121.27	3271.25	3472.97	1313.91	1396.50	474.295	483.170	494.025
TURBINE INLET		3070.41	3215.41	3410.63	1298.32	1378.28	470.156	478.805	489.376
TURBINE EXIT / INJECTOR INLET	PSIA	1964.01	1993.88	2031.38	957.883	982.234	377.881	381.726	386.260
TC INJECTOR FACE	PSIA	1837.87	1854.00	1872.80	918.935	936.400	367.574	370.800	374.560
CASE 1 LETS2 O2S STAGED COMBUSTION CYCLE					F=25K	MRE=6.50		15 JUL 71	14:14
CASE 2 LETS2 O2S STAGED COMBUSTION CYCLE					F=25K	MRE=6.00		15 JUL 71	14:13
CASE 3 LETS2 O2S STAGED COMBUSTION CYCLE					F=25K	MRE=5.50		15 JUL 71	14:15
CASE 4 LETS2 O2S STAGED COMBUSTION CYCLE					F=12.5K	MRE=6.50		15 JUL 71	14:17
CASE 5 LETS2 O2S STAGED COMBUSTION CYCLE					F=12.5K	MRE=5.50		15 JUL 71	14:17
CASE 6 LETS2 O2S STAGED COMBUSTION CYCLE					F=5.0K	MRE=6.50		15 JUL 71	14:19
CASE 7 LETS2 O2S STAGED COMBUSTION CYCLE					F=5.0K	MRE=6.00		15 JUL 71	14:19
CASE 8 LETS2 O2S STAGED COMBUSTION CYCLE					F=5.0K	MRE=5.50		15 JUL 71	14:20

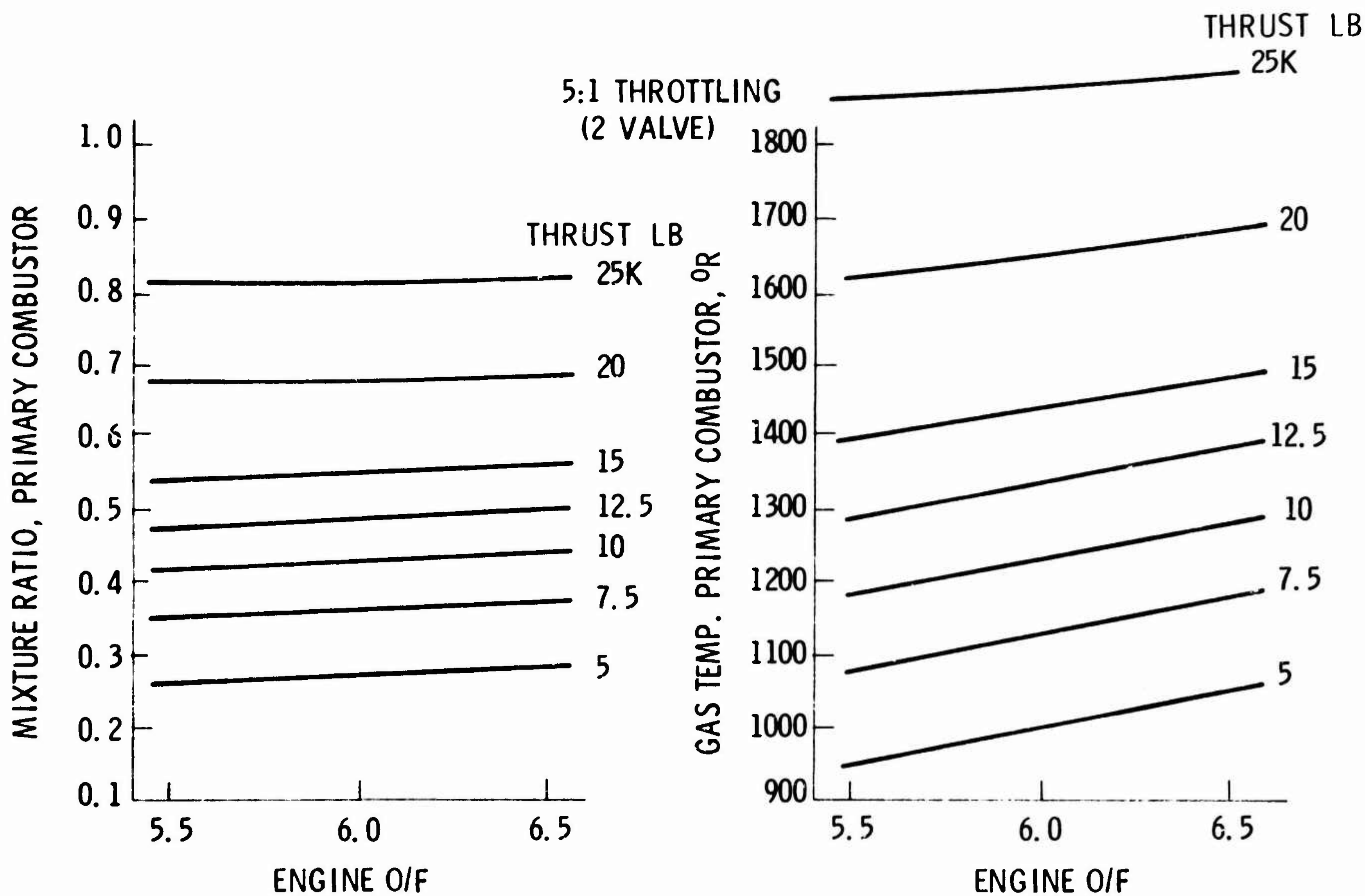


Figure 212. Preburner MR and T_{Ti} at Off Design Conditions, 5:1 Throttling (2 Valve)

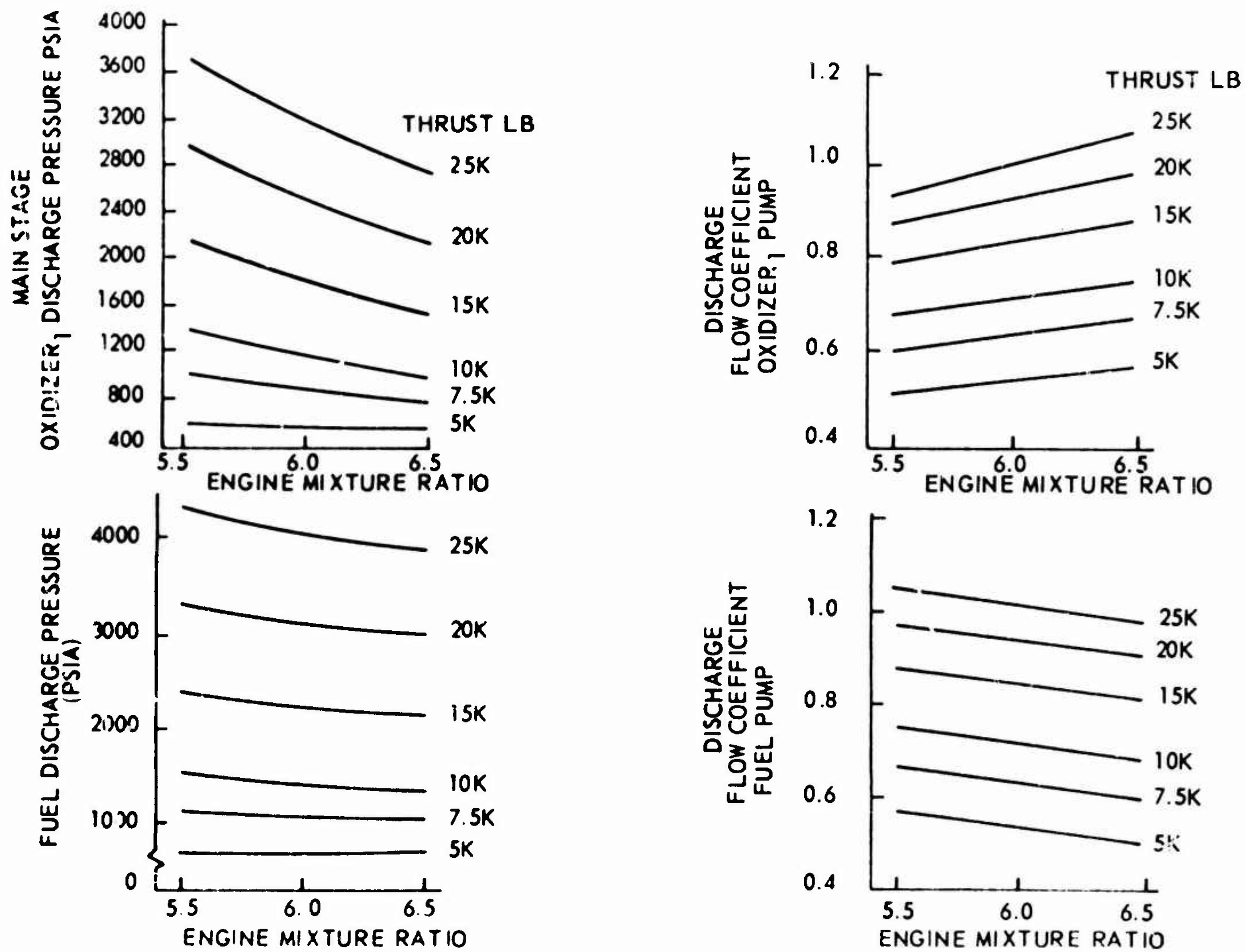


Figure 213. Fuel and Oxidizer Off-Design Operation (2 Valve)

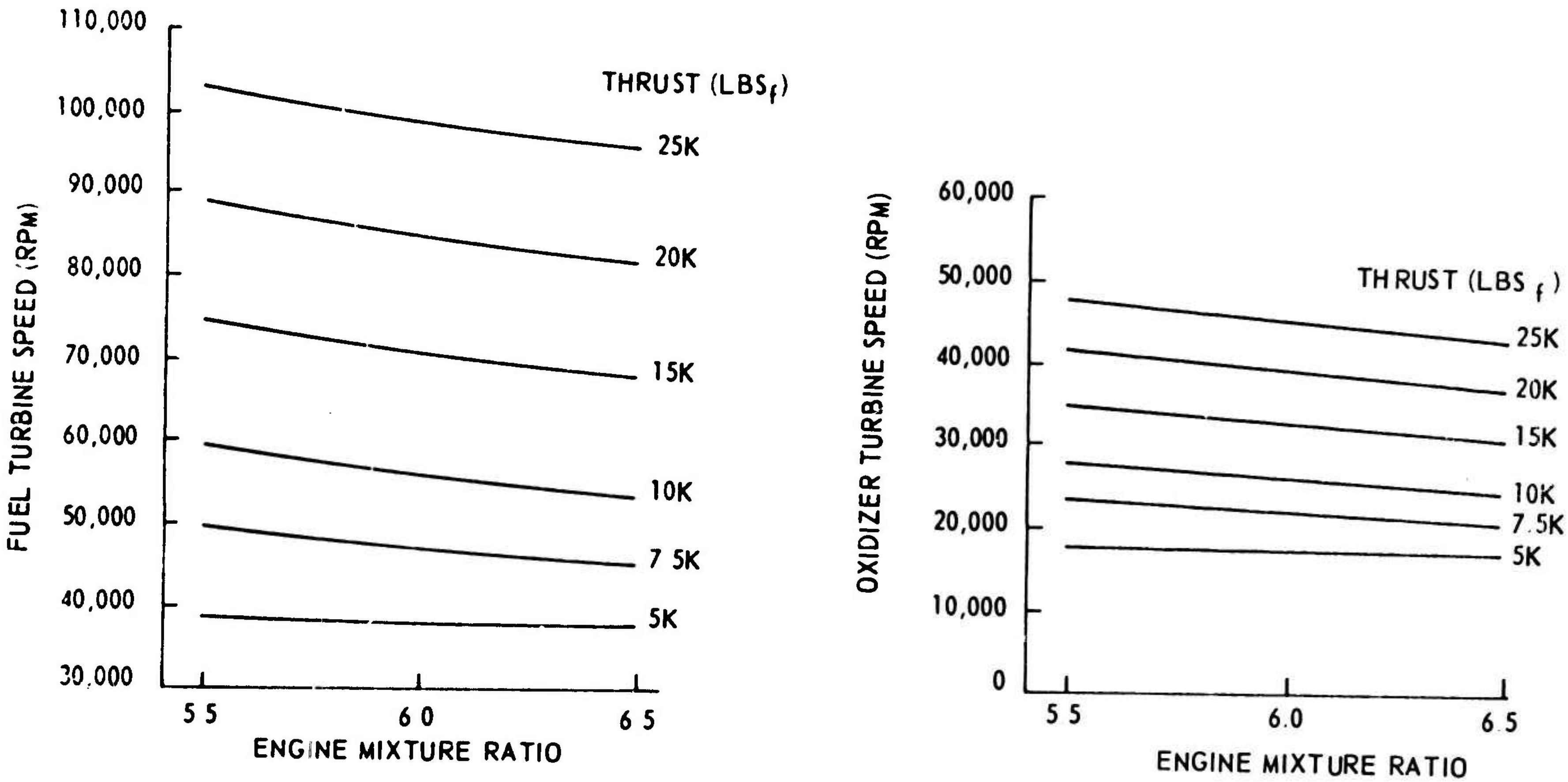


Figure 214. Fuel and Oxidizer Turbine Speed vs Mixture Ratio (2 Valve)

need for a third control valve. This valve bypasses warm hydrogen from the cooling jacket exit into the turbine exhaust. In conjunction with the preburner oxidizer valve, it makes possible control of turbine inlet temperature as well as thrust and mixture ratio. In the cases presented in Tables LVII and LVIII, a turbine temperature of 1660°R was held for thrust of 20K and less, however, in the final control system, it is probable that there will be enough range between the minimum allowable mixture ratio and maximum turbine temperature that no temperature feedback would be required. Figure 215 compares preburner mixture ratio and preburner injector stiffness for the two and three valve systems. Other parameters (pump speeds, flow coefficients, etc.) are only slightly different between the 2 and 3 valve systems and are presented in Figures 216 through 219.

A summary of the most significant extreme conditions for the 3-valve engine over a 5:1 throttling range is:

Cooling Jacket -
- Maximum pressure drop 912 psi at 5.5, 25K
- Maximum bulk temperature 478°R at 6.5, 5K

Gas Injector -
- Minimum stiffness 4% at 6.5, 5K

Main Oxidizer Injector -
- Minimum stiffness 20% at 6.0, 15K

Preburner -
- Maximum mixture ratio 0.82 at 6.5, 25K
- Minimum mixture ratio 0.64 at 6.5, 5K

Preburner Oxidizer Injector -
- Main stiffness 7.1% at 6.5, 12.5K

Preburner Fuel Injector -
- Minimum stiffness 6.5% at 5.5, 5K

Fuel Boost Pump -
- Maximum flow coefficient 105% at 5.5, 25K
- Minimum flow coefficient 45% at 6.5, 5K

Fuel Turbine -
- Maximum speed 24470 rpm at 5.5, 25K

Main Fuel Pump -
- Maximum flow coefficient 104% at 5.5, 25K
- Minimum flow coefficient 52% at 6.5, 5K
- Maximum discharge pressure (3rd stage) 4756 psia at 5.5, 25K

Oxidizer Boost Pump -
- Maximum flow coefficient 107% at 6.5, 25K
- Minimum flow coefficient 45% at 5.5, 5K

Oxidizer Turbine -
- Maximum speed 15460 rpm at 5.5, 25K

TABLE LVII

PERFORMANCE DATA, 10:1 THROTTLING AT CONSTANT MIXTURE RATIO

SUMMARY ENGINE PERFORMANCE DATA

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
ENGINE SPECIFIC IMPULSE	SEC	465.669	465.177	463.282	461.806	460.823	458.631	455.242	450.651
TOTAL OXIDIZER FLOW RATE	LB/SEC	46.0212	36.9384	27.8080	23.1887	18.5792	14.0022	9.41097	4.80902
TOTAL FUEL FLOW RATE	LB/SEC	7.67003	6.15610	4.63454	3.86472	3.09653	2.33371	1.56893	.801464
TOTAL FLOW LEAVING SYSTEM	LB/SEC	53.6913	43.0945	32.4425	27.0534	21.6757	16.3359	10.9799	5.61048
OX TANK OUTFLOW	LB/SEC	46.0209	36.9361	27.8086	23.1885	18.5807	14.0023	9.41146	4.80905
FUEL TANK OUTFLOW	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
OXIDIZER PRESSURE-INTERFACE	PSIA	46.3584	46.5333	46.6705	46.7252	46.7699	46.8045	46.8294	46.8446
OXIDIZER ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
OXIDIZER TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
OXIDIZER DENSITY	LB/FT3	71.0884	71.0881	71.0878	71.0877	71.0876	71.0875	71.0874	71.0874
FUEL PRESSURE-INTERFACE	PSIA	20.8096	20.9876	21.1273	21.1830	21.2284	21.2637	21.2891	21.3045
FUEL ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
FUEL TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
FUEL DENSITY	LB/FT3	4.34937	4.34957	4.34973	4.34979	4.34984	4.34988	4.34990	4.34992
OXID. BOOST PUMP SPEED	RPM	14462.4	12472.0	10475.5	9460.49	8472.50	7376.69	6137.30	4490.21
OXID. MAIN PUMP SPEED	RPM	49869.7	42398.5	34769.4	30958.1	27164.2	23040.8	18521.1	12807.1
FUEL BOOST PUMP SPEED	RPM	23150.6	19871.3	16740.7	15210.8	13637.9	11928.3	9952.16	7367.77
FUEL MAIN PUMP SPEED	RPM	80058.2	67822.3	55819.3	49952.5	43871.5	37342.0	30009.3	20970.9
OXID PUMP DISCH PRESS-STG1	PSIA	3105.69	2343.43	1659.04	1351.28	1068.20	793.438	535.162	276.174
OXID PUMP DISCH PRESS-STG2	PSIA	4665.91	3608.87	2560.17	2080.69	1637.80	1210.31	805.080	405.306
FUEL PUMP DISCH PRESS STG3	PSIA	4475.79	3374.28	2377.45	1927.23	1510.07	1111.47	734.451	372.874
PRIMARY COMBUSTOR PRESSURE	PSIA	3271.25	2409.02	1668.09	1338.02	1033.65	747.718	479.984	232.682
THRUST CHAMBER PRESSURE	PSIA	1800.00	1440.00	1080.00	900.000	720.000	540.000	360.000	181.825
OX. TC VALVE ADMITTANCE	KW	1.48840	1.45756	1.31839	1.34560	1.27839	1.18562	1.03474	.841244
OX. PB VALVE ADMITTANCE	KW	.198186	.126244	.938214-01	.776699-01	.652594-01	.509116-01	.376096-01	.229950-01
FU. CJ BYP. VALVE ADMITTANCE	KW								
FU. PB BYP. VALVE ADMITTANCE	KW		.234755-01	.216987	.332929	.461110	.629586	.864552	1.21392

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71 14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71 14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71 14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71 14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71 14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71 14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71 14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71 08:54

NOTE: Boost Pump = LPP

TABLE LVII (Cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   10:1 THROTTLEABLE   THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
THRUST CHAMBER PERFORMANCE DATA									
THRUST CHAMBER PRESSURE	PSIA	1800.00	1440.00	1080.00	900.000	720.000	540.000	360.000	181.825
TC INJECTOR MIXTURE RATIO	-	6.09369	6.09374	6.09324	6.09312	6.09289	6.09240	6.08994	6.09088
TC AREA RATIO	-	270.000	270.000	270.000	270.000	270.000	270.000	270.000	270.000
TC SPECIFIC IMPULSE	SEC	466.263	465.768	463.866	462.387	461.401	459.202	455.801	451.194
TC THRUST	LBS	24979.2	20028.1	15016.2	12481.9	9979.43	7485.22	4993.89	2526.03
DUMP COOLANT THRUST	LBS	23.1278	18.5022	13.8767	11.5639	9.25112	6.93834	4.62556	2.33623
DUMP COOLANT FLOW RATE	LB/SEC	.117999	.943992-01	.707994-01	.589995-01	.471996-01	.353997-01	.235998-01	.119196-01
REGEN COOLANT FLOW RATE	LB/SEC	7.33920	5.89854	4.44352	3.70544	2.97530	2.23928	1.51004	.768784
INLET PRESSURE	PSIA	4417.02	3336.36	2355.84	1912.19	1500.29	1105.89	731.915	372.222
INLET TEMPERATURE	R	121.592	101.459	84.1467	76.7424	70.3405	63.6988	56.9207	49.9582
INLET DENSITY	LB/FT3	4.06291	4.07343	4.06112	4.06066	4.03037	4.00706	4.00408	4.03844
INLET ENTHALPY	BTU/LB	238.928	152.892	79.5333	47.8341	19.5138	-8.29370	-35.9504	-64.1668
PRESSURE DROP	PSI	858.942	695.240	539.962	459.925	380.864	297.831	213.610	120.518
ENTHALPY RISE	BTU/LB	1279.33	1331.94	1407.33	1460.44	1524.99	1616.09	1744.24	2006.67
TEMPERATURE RISE	R	324.935	340.339	361.304	375.657	393.014	418.227	453.843	528.245
OUTLET PRESSURE	PSIA	3558.08	2641.12	1815.88	1452.27	1119.43	808.063	518.305	251.704
OUTLET TEMPERATURE	R	446.527	441.798	445.450	452.400	463.355	481.925	510.764	578.203
OUTLET DENSITY	LB/FT3	1.27556	.995774	.704921	.564021	.431950	.304020	.186718	.818191-01
OUTLET ENTHALPY	BTU/LB	1518.25	1484.83	1486.87	1508.27	1544.50	1607.79	1708.28	1942.50
GAS INJECTOR FLOW RATE	LB/SEC	13.5627	10.1169	7.26777	5.91251	4.62140	3.38350	2.19811	1.07208
INLET PRESSURE	PSIA	1993.68	1575.88	1174.69	975.746	778.208	581.532	386.444	194.428
PRESSURE DROP	PSI	139.885	92.6820	62.2912	48.7463	36.6084	25.3321	15.6444	7.14807
DP/PCFACE	-	.754503-01	.624879-01	.559972-01	.525850-01	.493641-01	.455450-01	.421909-01	.381678-01
TC OXID. INJECTOR FLOW RATE	LB/SEC	39.9087	32.8007	25.0459	21.0334	16.9706	12.8884	8.74135	4.51723
INLET PRESSURE	PSIA	2460.69	1893.45	1338.89	1134.23	911.240	688.630	471.722	250.617
INLET TEMPERATURE	R	184.158	179.020	174.630	172.416	170.118	169.345	167.368	165.689
INLET ENTHALPY	BTU/LB	-43.7952	-46.9804	-49.7828	-51.0151	-52.1076	-53.1958	-54.2472	-55.3716
OUTLET TEMPERATURE	R	552.825	464.444	419.916	402.842	390.696	414.346	418.890	476.030
OUTLET DENSITY	LB/FT3	11.1768	11.9567	10.0199	8.66387	6.94916	4.61826	2.96515	1.29118
PRESSURE DROP	PSI	612.692	410.251	226.488	207.234	169.640	132.430	100.922	63.3371
DP/PCFACE	-	.330470	.276598	.203603	.223554	.228749	.238099	.272172	.338195
TC IGNITER OXID. FLOW RATE	LB/SEC	.688548-01	.512743-01	.359293-01	.284062-01	.231452-01	.174051-01	.112575-01	.621636-02
FUEL FLOW RATE	LB/SEC	.101262	.795960-01	.585985-01	.484120-01	.384798-01	.287919-01	.175291-01	.974520-02
MIXTURE RATIO	-	.679965	.644182	.613144	.586760	.601489	.604512	.642217	.637889

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00 W/FPBYV, TTIT=1	15 JUL 71	14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	19 JUL 71	14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	19 JUL 71	14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	19 JUL 71	14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0	16 JUL 71	08:54

TABLE LVII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
PREBURNER PERFORMANCE DATA									
PREBURNER PRESSURE	PSIA	3271.25	2409.02	1668.09	1338.02	1033.65	747.718	479.984	232.682
MIXTURE RATIO	-	.811066	.693848	.693927	.691162	.685482	.675849	.659235	.622810
GAS TEMPERATURE	R	1878.62	1676.02	1676.02	1676.99	1678.63	1681.30	1686.36	1689.92
SPECIFIC HT. RATIO	-	1.36418	1.37058	1.37081	1.37105	1.37143	1.37200	1.37290	1.37473
GAS CONSTANT	FT	422.473	451.952	451.931	452.676	454.215	456.849	461.465	471.923
FLOW RATE	LB/SEC	13.4942	9.97041	6.65607	5.20410	3.90259	2.71908	1.65797	.745402
PB OXID. INJECTOR FLOW RATE	LB/SEC	5.95248	4.01279	2.67081	2.08082	1.55099	1.06898	.640095	.277010
INLET PRESSURE	PSIA	3847.98	2697.28	1830.99	1435.67	1130.80	813.307	546.021	275.986
INLET TEMPERATURE	R	206.445	196.941	188.298	184.530	181.299	177.134	174.468	170.102
INLET DENSITY	LB/FT3	67.4570	67.7505	68.1223	68.3982	68.7195	68.2624	68.5615	69.3915
TEMPERATURE RISE	R	205.124	184.212	171.568	164.623	158.349	191.580	285.753	398.344
ENTHALPY RISE	BTU/LB	83.7229	82.1259	87.4259	91.1312	96.9574	115.137	145.539	175.282
OUTLET TEMPERATURE	R	411.569	381.153	359.866	349.153	339.648	368.714	460.220	568.446
OUTLET DENSITY	LB/FT3	30.6872	28.0971	21.9928	18.2004	13.4201	7.54792	3.32163	1.19631
PRESSURE DROP	PSI	576.726	288.256	162.905	97.6498	97.1502	65.5886	66.0367	43.3041
ΔP/PCPB	-	.176301	.119657	.976598-01	.729811-01	.939878-01	.877184-01	.137581	.186108
PB FUEL INJECTOR FLOW RATE	LB/SEC	7.33825	5.80080	3.86750	3.02719	2.28163	1.59231	.982282	.449782
INLET PRESSURE	PSIA	3538.27	2624.73	1802.74	1440.85	1109.82	800.327	512.577	248.316
INLET TEMPERATURE	R	446.527	441.798	445.450	452.400	463.355	481.925	510.764	578.203
INLET DENSITY	LB/FT3	1.26919	.990096	.700084	.559865	.428330	.301203	.184667	.807260-01
INLET ENTHALPY	BTU/LB	1516.25	1484.83	1486.67	1508.27	1544.50	1607.79	1708.28	1942.50
PRESSURE DROP	PSI	267.024	215.710	134.653	102.834	76.1693	52.6095	32.5933	15.6334
ΔP/PCPB	-	.816275-01	.895425-01	.807231-01	.768553-01	.736898-01	.703600-01	.679051-01	.671878-01
PB IGNITER OXID. FLOW RATE	LB/SEC	.908130-01	.718615-01	.554976-01	.460379-01	.361929-01	.271823-01	.186904-01	.937757-02
FUEL FLOW RATE	LB/SEC	.111696	.335346-01	.618211-01	.518891-01	.357981-01	.302180-01	.173931-01	.110396-01
MIXTURE RATIO	-	.813034	.360261	.897713	.887236	1.01103	.899540	1.07458	.849450

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71 14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71 14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71 14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71 14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71 14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71 14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71 14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71 08:54

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TABLE LVII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 10:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
FUEL BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	19.3069	20.0197	20.5786	20.8015	20.9835	21.1246	21.2262	21.2881
ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
TEMPERATURE	R	37.1297	37.6857	37.7954	37.7934	37.7919	37.7906	37.7898	37.7892
SHAFT SPEED	RPM	23150.6	19871.3	16740.7	15210.8	13637.9	11928.3	9932.16	7367.77
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
VOLUME FLOW RATE	GPM	791.766	635.351	478.276	398.794	319.534	240.790	161.840	82.6954
HEAD RISE (ACTUAL)	FT	906.165	716.881	559.856	490.024	421.098	344.423	257.497	154.281
PRESSURE RISE	PSI	28.5951	22.5719	17.5951	15.3876	13.2121	10.7960	8.06227	4.82406
(Q/N)/(QD/ND)	-	.987155	.922863	.824623	.756743	.676273	.582652	.470320	.323964
SPECIFIC SPEED RPM-GPM**.5/FT**.75		3944.17	3615.34	3180.95	2916.50	2622.51	2315.15	1965.66	1530.53
EFFICIENCY	-	.709794	.699485	.683187	.665555	.638851	.595920	.528139	.416674
NPSH (NO TSH)	FT	40.7396	65.1842	84.6464	92.1907	98.3524	103.128	106.568	108.663
NPSP (NO TSH)	PSI	1.25003	1.96842	2.55651	2.78452	2.97076	3.11511	3.21912	3.28244
THERMODYNAMIC SUCT. HEAD	FT	120.569	126.096	127.186	127.167	127.151	127.139	127.130	127.125
SUCTION SPECIFIC SPEED		14391.9	9738.27	6593.56	5329.19	4189.29	3131.31	2113.95	1113.49
S/SDESIGN	-	.319819	.216406	.146524	.118427	.930953-01	.695847-01	.469767-01	.247442-01
S/SMAX AT Q/QD	-	.323109	.218813	.154064	.132118	.116326	.106653	.975842-01	.581484-01
SHAFT HORSEPOWER	HP	17.6040	11.4711	6.90560	5.17360	3.71135	2.45238	1.39049	.539586
DISCHARGE PRESSURE	PSIA	28.5951	22.5719	17.5951	15.3876	13.2121	10.7960	8.06227	4.82406
ENTHALPY	BTU/LB	-104.619	-104.943	-105.207	-105.314	-105.413	-105.517	-105.633	-105.784
TEMPERATURE	R	38.1035	38.0403	37.9905	37.9731	37.9591	37.9468	37.9345	37.9158
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
INLET PRESSURE	PSIA	198.118	160.315	126.435	109.949	94.1869	78.0299	61.1077	43.0280
ENTHALPY	BTU/LB	-96.0091	-98.3034	-100.216	-101.021	-101.784	-102.562	-103.378	-104.317
TEMPERATURE	R	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337
PRESSURE DROP	PSI	38.7661	31.0805	24.8134	22.2699	19.9250	17.4514	14.7053	11.1569
ISEN. SPOUTING VELOCITY	FT/SEC	328.092	293.618	262.935	250.042	238.243	225.926	212.950	197.257
MEAN WHEEL SPEED	FT/SEC	205.725	176.584	148.764	135.169	121.191	106.000	88.2608	65.4727
VELOCITY RATIO - U/CO	-	.627031	.601406	.565782	.540582	.508686	.469178	.414465	.331914
EFFICIENCY	-	.763447	.764996	.762512	.757498	.747281	.728632	.691851	.612276
SHAFT HORSEPOWER	HP	17.8106	11.4717	6.90356	5.17167	3.71139	2.45239	1.39052	.539541
OUTLET PRESSURE	PSIA	159.350	129.234	101.622	87.6792	74.2619	60.5786	46.4024	31.8710
ENTHALPY	BTU/LB	-97.6503	-99.6204	-101.269	-101.967	-102.631	-103.305	-104.005	-104.792
TEMPERATURE	R	39.7519	39.2223	38.8354	38.7068	38.5918	38.4817	38.3726	38.2389
TURBINE TORQUE	FT-LBS	4.04063	3.03203	2.16587	1.78572	1.42930	1.07980	.735305	.384612

CASE 1 LETS2 O.S STAGED COMBUSTION CYCLE F=25K MRE=6.00 15 JUL 71 14:13
CASE 2 LETS2 O.S STAGED COMBUSTION CYCLE F=20K MRE=6.00 W/FPBYV, TTIT=1 15 JUL 71 14:21
CASE 3 LETS2 O.S STAGED COMBUSTION CYCLE F=15K MRE=6.00 15 JUL 71 14:56
CASE 4 LETS2 O.S STAGED COMBUSTION CYCLE F=12.5K MRE=6.00 15 JUL 71 14:56
CASE 5 LETS2 O.S STAGED COMBUSTION CYCLE F=10K MRE=6.00 19 JUL 71 14:42
CASE 6 LETS2 O.S STAGED COMBUSTION CYCLE F=7.5K MRE=6.00 19 JUL 71 14:43
CASE 7 LETS2 O.S STAGED COMBUSTION CYCLE F=5.0K MRE=6.00 19 JUL 71 14:43
CASE 8 LETS2 O.S STAGED COMBUSTION CYCLE F=2.5K MRE=6.0 16 JUL 71 08:54

TABLE LVII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
FUEL PUMP - HIGH SPEED INDUCER AND FIRST STAGE									
HIGH SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	47.9020	42.5916	38.1738	36.1891	34.1956	31.9206	29.2885	26.1122
ENTHALPY	BTU/LB	-104.619	-104.943	-105.207	-105.314	-105.413	-105.517	-105.633	-105.784
TEMPERATURE	R	38.0924	38.0333	37.9867	37.9705	37.9574	37.9459	37.9341	37.9157
SHAFT SPEED	RPM	80058.2	67822.3	55819.3	49952.5	43871.5	37342.0	30009.3	20970.9
WEIGHT FLOW RATE	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
VOLUME FLOW RATE	GPM	791.302	635.122	478.198	398.780	319.571	240.863	161.925	82.7562
HEAD RISE (ACTUAL)	FT	4781.13	3615.30	2677.64	2255.52	1849.01	1432.02	996.285	534.478
PRESSURE RISE	PSI	150.216	117.723	88.2614	73.7600	59.9913	46.1094	31.8192	16.9158
(Q/N)/(QD/ND)	-	.998392	.945909	.865343	.806381	.735783	.651535	.545034	.398610
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3916.78	3666.00	3279.25	3047.82	2781.39	2489.56	2153.40	1716.21
EFFICIENCY	-	.713557	.699712	.689426	.675176	.654729	.622663	.567703	.468033
NPSH (NO TSH)	FT	961.897	791.670	649.775	585.607	520.861	446.665	360.648	257.173
NPSP (NO TSH)	PSI	29.0590	23.9154	19.6278	17.6883	15.7310	13.4880	10.8884	7.76290
THERMODYNAMIC SUCT. HEAD	FT	130.139	129.551	129.088	128.927	128.797	128.682	128.565	128.382
SUCTION SPECIFIC SPEED		11854.9	10221.8	8279.28	7217.84	6094.70	4933.28	3671.04	2192.57
S/SDESIGN	-	.395165	.340727	.275976	.240595	.203157	.164443	.122368	.730856-01
S/SMAX AT Q/QD	-	.399893	.343701	.283787	.256299	.232291	.215139	.208008	.150631
SHAFT HORSEPOWER	HP	93.4423	57.8313	32.7287	23.4741	15.9010	9.75841	5.00502	1.66418
DISCHARGE PRESSURE	PSIA	198.118	160.315	126.435	109.949	94.1869	78.0299	61.1077	43.0280
ENTHALPY	BTU/LB	-96.0091	-98.3034	-100.216	-101.021	-101.784	-102.562	-103.378	-104.317
TEMPERATURE	R	40.0816	39.4534	38.9971	38.8429	38.7059	38.5747	38.4449	38.2886
FUEL PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
VOLUME FLOW RATE (1)	GPM	921.564	726.082	541.374	451.269	362.357	274.610	186.988	98.8457
HEAD RISE (ACTUAL)	FT	49605.1	36918.5	25824.7	20951.1	16362.8	12026.0	7867.13	3872.16
PRESSURE RISE	PSI	1466.79	1093.79	775.167	625.771	488.244	362.387	230.384	111.684
(Q/N)/(QD/ND)		1.00644	.936013	.847973	.789853	.722142	.642964	.544788	.412106
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	731.179	686.171	637.539	609.354	577.240	538.845	491.129	424.748
EFFICIENCY	-	.606118	.599149	.585309	.572992	.548007	.519059	.474730	.393416
NPSH (NO TSH)	FT	3559.15	2878.87	2161.86	1763.22	1388.13	1037.30	663.594	282.377
NPSP (NO TSH)	PSI	98.7807	81.6876	62.3377	51.1389	40.4833	30.4197	19.5732	8.38725
THERMODYNAMIC SUCT. HEAD	FT	219.274	197.960	181.605	175.396	169.164	161.303	153.396	144.579
SUCTION SPECIFIC SPEED		5042.96	4423.72	3856.02	3632.09	3368.78	3037.74	2685.35	2219.77
S/SDESIGN	-	.504296	.442372	.385602	.363209	.336878	.303774	.268535	.221977
S/SMAX AT Q/QD	-	.507998	.432353	.401754	.402459	.410098	.425653	.452783	.442601
SHAFT HORSEPOWER	HP	1221.14	740.587	401.806	279.181	184.078	108.848	53.3202	16.8576
DISCHARGE PRESSURE	PSIA	1626.14	1223.02	876.789	713.450	562.506	422.966	276.786	143.555
ENTHALPY	BTU/LB	28.2682	-4.77645	-33.0474	-45.1300	-55.9967	-66.8439	-77.6413	-88.8956
TEMPERATURE	R	72.3663	64.0803	57.0414	54.0573	51.6643	48.7757	46.2710	43.2059

(1) VOLUME FLOW INCLUDES 7% RECIRCULATION

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71 14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71 14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71 14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71 14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71 14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71 14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71 14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71 08:54

TABLE LVII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 10:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4996.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99835	6.00029
FUEL PUMP - SECOND AND THIRD STAGES									
SECOND STAGE FUEL PUMP									
SUCTION PRESSURE	PSIA	1620.14	1223.02	876.789	713.450	562.506	422.966	276.786	143.555
ENTHALPY	BTU/LB	28.2682	-4.77645	-33.0474	-45.1300	-55.9967	-66.8439	-77.6413	-88.8956
TEMPERATURE	R	72.3447	64.0662	57.0335	54.0526	51.6594	48.7739	46.2697	43.2058
SHAFT SPEED	RPM	80058.2	67822.3	55819.3	49952.5	43871.5	37342.0	30009.3	20970.9
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.55858	.801509
VOLUME FLOW RATE (1)	GPM	913.890	728.404	546.423	457.881	368.952	280.114	191.182	100.955
HEAD RISE (ACTUAL)	FT	49760.6	36850.2	25759.5	20902.8	16321.2	11996.6	7860.64	3871.11
PRESSURE RISE	PSI	1449.36	1081.92	754.908	611.045	477.895	346.777	231.773	118.083
(Q/N)/(QD/ND)	-	.999153	.940034	.856818	.802303	.736091	.656569	.557617	.421360
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	726.422	688.223	641.720	614.866	583.585	545.220	497.034	429.343
EFFICIENCY	-	.607008	.599308	.586089	.576264	.552054	.522961	.480681	.398884
SHAFT HORSEPOWER	HP	1223.18	739.020	400.259	276.956	182.263	107.771	52.6166	16.6220
DISCHARGE PRESSURE	PSIA	3075.50	2304.94	1631.70	1324.50	1040.40	769.743	508.559	261.638
ENTHALPY	BTU/LB	133.612	74.2381	23.4323	1.48232	-18.0052	-37.3654	-56.6269	-76.4244
TEMPERATURE	R	97.2033	82.9350	70.7983	65.5267	61.1393	56.2341	51.6870	46.7862
THIRD STAGE FUEL PUMP									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.67015	6.15602	4.63476	3.86475	3.09678	2.33371	1.56858	.801509
VOLUME FLOW RATE (1)	GPM	905.296	727.810	550.492	461.575	374.094	285.451	194.814	102.746
HEAD RISE (ACTUAL)	FT	49747.2	36700.7	25640.6	20841.2	16268.3	11955.1	7848.40	3869.56
PRESSURE RISE	PSI	1400.29	1069.34	745.755	602.734	469.665	341.724	225.892	111.236
(Q/N)/(QD/ND)	-	.999600	.948607	.871783	.816819	.753771	.675732	.573860	.433099
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	723.144	690.043	646.344	618.709	589.067	551.821	502.319	433.264
EFFICIENCY	-	.607005	.599611	.587320	.577796	.557199	.528446	.487779	.405669
SHAFT HORSEPOWER	HP	1222.85	735.649	397.576	275.408	179.996	106.284	51.7702	16.3374
FUEL TURBINE SPEED	RPM	80058.2	67822.3	55819.3	49952.5	43871.5	37342.0	30009.3	20970.9
COMBINED FUEL PUMP PERFORMANCE BASED ON DELIVERED FLOW									
TOTAL FUEL TPA POWER REQ'D	HP	3760.62	2273.09	1232.37	855.019	562.238	332.661	162.712	51.4812
TOTAL FUEL TPA HEAD RISE	FT	153694.	114085.	79902.5	64950.6	50801.3	37409.7	24572.5	12147.3
OVERALL PUMP EFFICIENCY	-	.570695	.561756	.546366	.533785	.508748	.477164	.430697	.343856

(1) VOL. FLOW INCLUDES 7% RECIRC.

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71 14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71 14:21
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71 14:56
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71 14:56
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71 14:42
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71 14:43
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71 14:43
CASE 8	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71 08:54

TABLE LVII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
OXIDIZER BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	44.8915	45.5884	46.1349	46.3528	46.5307	46.6687	46.7681	46.8286
ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
TEMPERATURE	R	162.927	162.922	162.918	162.915	162.910	162.907	162.904	162.903
SHAFT SPEED	RPM	14462.4	12472.0	10475.5	9460.49	8472.50	7376.69	6137.30	4490.21
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8086	23.1885	18.5807	14.0023	9.41146	4.80905
VOLUME FLOW RATE	GPM	290.529	233.182	175.562	146.395	117.305	88.4008	59.4177	30.3612
HEAD RISE (ACTUAL)	FT	246.882	198.034	153.252	131.994	113.232	91.6778	68.4998	39.9091
PRESSURE RISE	PSI	121.878	97.7628	75.6553	65.1606	55.8983	45.2579	33.8158	19.7016
(G/N)/(GD/ND)	-	.989884	.921282	.825823	.762513	.682245	.590512	.477060	.333187
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3957.92	3607.69	3186.66	2939.42	2643.59	2340.95	1986.86	1558.20
EFFICIENCY	-	.660592	.649865	.633825	.617796	.593093	.554069	.491031	.389781
NPSH (NO TSH)	FT	60.2236	61.6441	62.7580	63.2051	63.5727	63.8576	64.0628	64.1878
NPSP (NO TSH)	PSI	29.7319	30.4325	30.9819	31.2025	31.3837	31.5243	31.6255	31.6871
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		10865.9	8258.23	5943.11	4876.69	3893.52	2933.53	1996.42	1042.67
S/SDESIGN	-	.241464	.183516	.132069	.108371	.865226-01	.651896-01	.443650-01	.231704-01
S/SMAX AT Q/QD	-	.244034	.185604	.138756	.120150	.107010	.979094-01	.908349-01	.532790-01
SHAFT HORSEPOWER	HP	31.2715	20.4647	12.2251	9.00780	6.44976	4.21246	2.38712	.895258
DISCHARGE PRESSURE	PSIA	121.878	97.7628	75.6553	65.1606	55.8983	45.2579	33.8158	19.7016
ENTHALPY	BTU/LB	-56.5647	-56.6534	-56.7343	-56.7704	-56.7997	-56.8324	-56.8657	-56.9134
TEMPERATURE	R	163.119	163.069	163.022	163.036	163.088	162.999	163.011	163.039
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8086	23.1885	18.5807	14.0023	9.41146	4.80905
INLET PRESSURE	PSIA	785.917	615.773	469.045	400.385	339.106	273.926	207.546	133.243
ENTHALPY	BTU/LB	-54.1338	-54.7537	-55.3165	-55.5667	-55.7828	-56.0108	-56.2369	-56.5141
TEMPERATURE	R	165.309	165.309	165.309	165.309	165.309	165.309	165.309	165.309
PRESSURE DROP	PSI	184.164	149.684	118.794	104.962	93.7462	81.3078	68.5417	50.3792
ISEN. SPOUTING VELOCITY	FT/SEC	177.347	160.246	143.250	135.315	129.034	121.918	115.278	105.357
MEAN WHEEL SPEED	FT/SEC	107.233	92.4751	77.6721	70.1459	62.8203	54.6953	45.5057	33.2931
VELOCITY RATIO - U/CO	-	.604651	.577082	.542212	.518388	.486849	.448623	.394747	.316002
EFFICIENCY	-	.764954	.763884	.757904	.750846	.737793	.716306	.675476	.593609
SHAFT HORSEPOWER	HP	31.2852	20.4716	12.2204	9.00779	6.44922	4.21243	2.38705	.895343
OUTLET PRESSURE	PSIA	601.752	466.089	350.250	295.423	245.360	192.619	139.004	82.8640
ENTHALPY	BTU/LB	-54.6143	-55.1454	-55.6271	-55.8413	-56.0281	-56.2234	-56.4161	-56.6457
TEMPERATURE	R	166.225	165.560	164.639	164.528	164.062	163.771	163.686	163.521
TURBINE TORQUE	FT-LBS	11.3614	8.62085	6.12692	5.00079	3.99788	2.99920	2.04276	1.04727

CASE 1 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71	14:13
CASE 2 LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FP8YV, TTIT=1	15 JUL 71	14:21
CASE 3 LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71	14:56
CASE 4 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71	14:56
CASE 5 LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71	14:42
CASE 6 LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71	14:43
CASE 7 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71	14:43
CASE 8 LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71	08:54

TABLE LVII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   10:1 THROTTLEABLE   THROTTLED AT CONSTANT MIXTURE RATIO   PAGE 8

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
OXIDIZER PUMP - HIGH SPEED INDUCER AND MAIN STAGE									
HIGH SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	166.770	143.351	121.790	111.513	102.429	91.9266	80.5839	66.5302
ENTHALPY	BTU/LB	-56.5647	-56.6534	-56.7343	-56.7704	-56.7997	-56.8324	-56.8657	-56.9134
TEMPERATURE	R	163.120	163.085	163.015	163.050	163.080	162.994	163.013	163.039
SHAFT SPEED	RPM	49869.7	42398.5	34769.4	30958.1	27164.2	23040.8	18521.1	12807.1
WEIGHT FLOW RATE	LB/SEC	46.0209	36.9361	27.8086	23.1885	18.5807	14.0023	9.41146	4.80905
VOLUME FLOW RATE	GPM	291.432	233.809	175.985	146.686	117.461	88.5584	59.5152	30.3969
HEAD RISE (ACTUAL)	FT	1258.01	959.511	705.107	586.316	480.063	369.328	257.606	135.297
PRESSURE RISE	PSI	619.147	472.422	347.254	288.872	236.677	182.000	126.962	66.7129
(Q/N)/(QD/ND)	-	1.00756	.950785	.872670	.816934	.745537	.662681	.554032	.409213
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4030.34	3760.49	3370.89	3146.81	2870.58	2573.67	2222.10	1779.92
EFFICIENCY	-	.665005	.649057	.639077	.625923	.606672	.577643	.526414	.435368
NPSH (NO TSH)	FT	307.759	260.126	216.384	195.383	176.783	155.681	132.618	104.014
NPSP (NO TSH)	PSI	151.467	128.075	106.566	96.2630	87.1565	76.7178	65.3612	51.2878
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		11474.7	9895.18	8064.01	7066.49	5971.37	4826.93	3575.61	2107.44
S/SDESIGN	-	.382490	.329839	.268800	.235550	.199046	.160898	.119187	.702481-01
S/SMAX AT Q/QD	-	.400133	.332652	.275564	.248977	.224908	.206045	.197489	.148288
SHAFT HORSEPOWER	HP	158.289	99.2786	55.7850	39.4931	26.7327	16.2775	8.37379	2.71725
DISCHARGE PRESSURE	PSIA	785.917	615.773	469.045	400.385	339.106	273.926	207.546	133.243
ENTHALPY	BTU/LB	-54.1338	-54.7537	-55.3165	-55.5667	-55.7828	-56.0108	-56.2369	-56.5141
TEMPERATURE	R	167.010	165.829	165.111	164.674	164.338	164.112	163.641	163.507
OXIDIZER PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	46.0209	36.9361	27.8086	23.1885	18.5807	14.0023	9.41146	4.80905
VOLUME FLOW RATE (1)	GPM	307.868	247.361	187.106	156.382	126.006	95.6587	65.1103	34.1998
HEAD RISE (ACTUAL)	FT	5105.12	3823.34	2667.52	2149.62	1675.58	1223.10	805.769	392.773
PRESSURE RISE	PSI	2503.96	1877.34	1308.79	1055.85	822.841	600.820	396.157	193.310
(Q/N)/(QD/ND)		1.01544	.959571	.885090	.830824	.762939	.682847	.578203	.439206
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	1446.87	1371.47	1281.33	1226.30	1164.31	1089.59	988.172	848.902
EFFICIENCY	-	.635209	.630706	.615331	.601096	.577720	.547219	.504963	.420742
NPSH (NO TSH)	FT	1192.19	914.713	680.339	567.989	466.876	359.753	250.488	136.449
NPSP (NO TSH)	PSI	583.075	448.169	333.267	278.612	228.997	176.546	123.061	67.1251
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		4302.12	3996.06	3554.58	3310.00	3016.55	2705.54	2345.52	1835.80
S/SDESIGN	-	.430212	.399606	.355458	.331000	.301655	.270554	.234552	.183580
S/SMAX AT Q/QD	-	.436299	.388011	.359057	.350581	.345950	.351845	.373371	.355268
SHAFT HORSEPOWER	HP	706.006	428.599	231.794	160.034	104.571	61.2101	29.7770	9.16243
DISCHARGE PRESSURE	PSIA	3105.69	2343.43	1659.04	1351.28	1068.20	793.438	535.162	276.174
ENTHALPY	BTU/LB	-43.7952	-46.9804	-49.7828	-51.0151	-52.1076	-53.1958	-54.2472	-55.3716
TEMPERATURE	R	184.158	179.020	174.630	172.416	170.118	169.345	167.368	165.689

(1) VOLUME FLOW INCLUDES 5% RECIRCULATION.

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00	15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00 W/FPBYV, TTTT=1	15 JUL 71	14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00	15 JUL 71	14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00	15 JUL 71	14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00	19 JUL 71	14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00	19 JUL 71	14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00	19 JUL 71	14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0	16 JUL 71	08:54

TABLE LVII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 10:1 THROTTLEABLE THROTTLED AT CONSTANT MIXTURE RATIO PAGE 9

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
OXIDIZER PUMP - HALF STAGE									
SUCTION PRESSURE	PSIA	2646.36	2224.99	1606.29	1319.14	1050.25	784.902	532.072	275.592
ENTHALPY	BTU/LB	-43.7952	-46.9804	-49.7828	-51.0151	-52.1076	-53.1958	-54.2472	-55.3716
TEMPERATURE	R	186.539	180.663	175.792	173.282	170.859	169.986	167.919	166.061
SHAFT SPEED	RPM	49869.7	42398.5	34769.4	30958.1	27164.2	23040.8	18521.1	12807.1
WEIGHT FLOW RATE	LB/SEC	6.04331	4.08416	2.72670	2.12687	1.58717	1.09657	.658731	.286074
VOLUME FLOW RATE	GPM	40.8821	27.9060	19.8514	14.8817	11.3045	7.95766	4.98791	2.36152
HEAD RISE (ACTUAL)	FT	3713.59	2824.21	1945.21	1554.85	1203.33	867.994	558.707	264.959
PRESSURE RISE	PSI	1619.55	1383.88	953.877	761.547	587.552	425.403	273.008	129.713
(G/N)/(GD/ND)	-	.931567	.747936	.610116	.546256	.472904	.392469	.306034	.209535
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	670.282	578.133	515.401	482.319	447.027	406.446	359.946	299.684
EFFICIENCY	-	.454885	.428320	.391436	.365823	.333544	.293324	.247619	.185707
SHAFT HORSEPOWER	HP	94.5862	52.3178	26.7099	18.0150	11.5870	6.71806	3.20384	.961381
DISCHARGE PRESSURE	PSIA	4665.91	3608.87	2560.17	2080.69	1637.80	1210.31	805.080	405.306
ENTHALPY	BTU/LB	-32.7854	-38.0416	-43.0177	-45.2139	-47.1690	-49.1319	-51.1308	-53.3870
TEMPERATURE	R	206.445	196.941	188.298	184.530	181.299	177.134	174.468	170.102

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71	14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71	14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71	14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71	14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71	14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71	14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71	08:54

TABLE LVII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
GAS TURBINE PERFORMANCE DATA									
FUEL TPA SHAFT SPEED	RPM	80058.2	67822.3	55819.3	49952.5	43871.5	37342.0	30009.3	20970.9
TURBINE INLET TEMPERATURE	R	1860.66	1660.63	1659.68	1660.04	1660.75	1660.38	1660.63	1660.67
WEIGHT FLOW RATE (1)	LB/SEC	10.0318	7.42170	4.94812	3.86892	2.90149	2.02151	1.23266	.554177
MEAN WHEEL SPEED	FT/SEC	1299.69	1101.22	906.329	811.071	712.334	606.317	487.256	340.501
ISEN. SPOUTING VELOCITY	FT/SEC	4763.78	4324.51	3946.07	3755.44	3574.24	3379.32	3160.79	2916.20
STAGE VEL. RATIO U/C	-	.385895	.360123	.324815	.305430	.281847	.253737	.218010	.165126
INLET PRESSURE (TOTAL)	PSIA	3215.41	2372.30	1646.39	1322.16	1022.53	740.539	475.981	231.064
EXIT PRESSURE (STATIC)	PSIA	1993.88	1575.88	1174.69	975.746	779.208	581.532	386.444	194.428
PRESSURE RATIO	-	1.61264	1.50538	1.40155	1.35502	1.31395	1.27343	1.23169	1.18843
SHAFT HORSEPOWER	HP	3760.67	2272.80	1232.89	855.137	562.093	332.553	162.558	51.4678
SHAFT TORQUE	FT-LBS	246.727	176.004	116.004	89.9108	67.2915	46.7732	28.4504	12.8900
EXIT TEMPERATURE	R	1730.36	1559.73	1577.65	1587.36	1597.21	1606.68	1617.94	1631.17
EFFICIENCY	-	.628508	.623843	.608782	.596252	.576934	.548066	.502207	.415489
OXID TPA SHAFT SPEED	RPM	49869.7	42398.5	34769.4	30958.1	27164.2	23040.8	18521.1	12807.1
TURBINE INLET TEMPERATURE	R	1860.66	1660.63	1659.68	1660.04	1660.75	1660.38	1660.63	1660.67
WEIGHT FLOW RATE (1)	LB/SEC	3.46200	2.55779	1.70761	1.33517	1.00131	.697625	.425391	.191247
MEAN WHEEL SPEED	FT/SEC	1096.26	932.025	764.318	680.536	597.136	506.494	407.139	281.532
ISEN. SPOUTING VELOCITY	FT/SEC	4763.78	4324.51	3946.07	3755.44	3574.24	3379.32	3160.79	2916.20
VELOCITY RATIO - U/C	-	.230123	.215521	.193691	.181213	.167066	.149880	.128809	.965406-01
INLET PRESSURE (TOTAL)	PSIA	3215.41	2372.30	1646.39	1322.16	1022.53	740.539	475.981	231.064
EXIT PRESSURE (STATIC)	PSIA	1993.88	1575.88	1174.69	975.746	778.208	581.532	386.444	194.428
PRESSURE RATIO	-	1.61264	1.50538	1.40155	1.35502	1.31395	1.27343	1.23169	1.18843
SHAFT HORSEPOWER	HP	959.195	579.986	314.419	217.665	142.957	84.3779	41.2154	12.9335
SHAFT TORQUE	FT-LBS	101.019	71.8457	47.4947	36.9274	27.6403	19.2338	11.6877	5.30394
EXIT TEMPERATURE	R	1764.36	1586.02	1599.06	1606.44	1613.93	1620.90	1629.26	1639.19
EFFICIENCY	-	.464497	.461301	.449882	.439781	.425183	.402952	.368966	.302547
TURBINE GAS PROPERTIES									
SPECIFIC HEAT RATIO	-	1.36418	1.37058	1.37081	1.37105	1.37143	1.37200	1.37290	1.37473
GAS CONSTANT	FT	422.473	451.952	451.931	452.676	454.215	456.849	461.465	471.923
MOLECULAR WEIGHT	-	3.65467	3.41629	3.41645	3.41083	3.39927	3.37967	3.34586	3.27172
TEMPERATURE	R	1860.66	1660.63	1659.68	1660.04	1660.75	1660.38	1660.63	1660.67

(1) INCLUDES 7% LEAKAGE FLOW

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71	14:13
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTTT=1	15 JUL 71	14:21
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71	14:56
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71	14:56
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71	14:42
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71	14:43
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71	14:43
CASE 8	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71	08:54

TABLE LVII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.37
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
CONTROLS COMPONENTS									
OXID. THRUST CHAMBER VALVE									
WEIGHT FLOW RATE	LB/SEC	39.9068	32.8007	25.0459	21.0332	16.9704	12.8883	8.74147	4.51676
INLET PRESSURE	PSIA	4105.69	2343.43	1659.04	1351.28	1068.20	793.438	535.162	276.174
INLET TEMPERATURE	R	184.158	179.020	174.630	172.416	170.118	169.345	167.368	165.689
INLET DENSITY	LB/FT3	70.5557	70.5608	70.6138	70.5293	70.3111	70.5743	70.3644	70.4967
PRESSURE DROP	PSI	635.843	447.849	318.920	216.170	156.393	104.481	63.2899	25.5168
3 ADMITTANCE	KA	1.48840	1.45756	1.31839	1.34560	1.27839	1.18562	1.03474	.841244
OXID. PREBURNER VALVE									
WEIGHT FLOW RATE	LB/SEC	5.95250	4.01250	2.67120	2.08083	1.55098	1.06939	.640041	.276697
INLET PRESSURE	PSIA	4665.91	3608.87	2560.17	2080.69	1637.80	1210.31	805.080	405.306
INLET TEMPERATURE	R	206.445	196.941	188.298	184.530	181.299	177.134	174.468	170.102
INLET DENSITY	LB/FT3	68.8206	69.1436	69.3682	69.4346	69.5182	69.3479	69.7596	69.8650
PRESSURE DROP	PSI	817.932	911.586	729.178	645.025	507.006	396.999	259.059	129.320
7 ADMITTANCE	KA	.196186	.126244	.938214-01	.776699-01	.652594-01	.509116-01	.376096-01	.229950-01
FUEL PREBURNER BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC		.957577-01	.575987	.680104	.695687	.647010	.528211	.320516
INLET PRESSURE	PSIA		2624.73	1802.74	1440.85	1109.82	800.327	512.577	248.316
INLET TEMPERATURE	R		441.798	445.450	452.400	463.355	481.925	510.764	578.203
INLET DENSITY	LB/FT3		.990096	.700084	.559865	.428330	.301203	.184667	.807260-01
PRESSURE DROP	PSI		1048.85	628.049	465.103	331.608	218.795	126.133	53.8875
ADMITTANCE	KA		.234755-01	.216987	.332929	.461110	.629586	.864552	1.21392
FUEL COOL. JKT. BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC								
INLET PRESSURE	PSIA	4417.02	3336.36	2355.84	1912.19	1500.29	1105.89	731.915	372.222
INLET TEMPERATURE	R	121.592	101.459	84.1467	76.7424	70.3405	63.6988	56.9207	49.9582
INLET DENSITY	LB/FT3	4.06291	4.07343	4.06112	4.06066	4.03037	4.00706	4.00408	4.03844
PRESSURE DROP	PSI	2423.14	1760.48	1181.15	936.445	722.083	524.362	345.470	177.794
ADMITTANCE	KA								
FUEL SHUTOFF VALVE									
WEIGHT FLOW RATE	LB/SEC	7.45720	5.99288	4.51434	3.76445	3.02250	2.27470	1.53365	.780724
INLET PRESSURE	PSIA	4475.79	3374.28	2377.45	1927.23	1510.07	1111.47	734.451	372.874
INLET TEMPERATURE	R	121.592	101.459	84.1467	76.7424	70.3405	63.6988	56.9207	49.9582
INLET DENSITY	LB/FT3	4.08897	4.09308	4.07464	4.07224	4.03907	4.01232	4.00668	4.03923
PRESSURE DROP	PSI	58.7699	37.9175	21.6131	15.0379	9.77393	5.57276	2.53681	.652100
ADMITTANCE	KA	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.00		15 JUL 71	14:13
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=20K	MRE=6.00	W/FPBYV, TTIT=1	15 JUL 71	14:21
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=15K	MRE=6.00		15 JUL 71	14:56
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.00		15 JUL 71	14:56
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=10K	MRE=6.00		19 JUL 71	14:42
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=7.5K	MRE=6.00		19 JUL 71	14:43
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.00		19 JUL 71	14:43
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.0		16 JUL 71	08:54

TABLE LVII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   10:1 THROTTLEABLE   THROTTLED AT CONSTANT MIXTURE RATIO

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25002.4	20046.6	15030.0	12493.4	9988.69	7492.16	4998.51	2528.57
ENGINE MIXTURE RATIO	-	6.00014	6.00030	6.00016	6.00010	6.00002	5.99999	5.99833	6.00029
SUMMARY FUEL PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100
TANK OUTLET TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
TANK OUTLET ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.263	-106.260
BOOST PUMP SUCTION	PSIA	19.3069	20.0197	20.5786	20.8015	20.9835	21.1246	21.2262	21.2881
DISCHARGE	PSIA	47.9020	42.5916	38.1738	36.1891	34.1956	31.9206	29.2885	26.1122
HIGH SPEED INDUCER SUCTION	PSIA	47.9020	42.5916	38.1738	36.1891	34.1956	31.9206	29.2885	26.1122
DISCHARGE	PSIA	198.118	160.315	126.435	109.949	94.1869	78.0299	61.1077	43.0280
BOOST TURBINE INLET	PSIA	198.118	160.315	126.435	109.949	94.1869	78.0299	61.1077	43.0280
DISCHARGE	PSIA	159.350	129.234	101.622	87.6792	74.2619	60.5786	46.4024	31.8710
1ST STAGE PUMP SUCTION	PSIA	159.350	129.234	101.622	87.6792	74.2619	60.5786	46.4024	31.8710
DISCHARGE	PSIA	1626.14	1223.02	876.789	713.450	562.506	422.966	276.786	143.555
2ND STAGE PUMP DISCHARGE	PSIA	3075.50	2304.94	1631.70	1324.50	1040.40	769.743	508.559	261.638
3RD STAGE PUMP DISCHARGE	PSIA	4475.79	3374.28	2377.45	1927.23	1510.07	1111.47	734.451	372.874
COOLING JACKET INLET	PSIA	4417.02	3336.36	2355.84	1912.19	1500.29	1105.89	731.915	372.222
DISCHARGE	PSIA	3558.08	2641.12	1815.88	1452.27	1119.43	808.063	518.305	251.704
PREBURNER INJECTOR INLET	PSIA	3536.27	2624.73	1802.74	1440.85	1109.82	800.327	512.577	248.316
SUMMARY OXIDIZER PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500
TANK OUTLET TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
TANK OUTLET ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
BOOST PUMP SUCTION	PSIA	44.8915	45.5884	46.1349	46.3528	46.5307	46.6687	46.7681	46.8286
DISCHARGE	PSIA	166.770	143.351	121.790	111.513	102.429	91.9266	80.5839	66.5302
HIGH SPEED INDUCER SUCTION	PSIA	166.770	143.351	121.790	111.513	102.429	91.9266	80.5839	66.5302
DISCHARGE	PSIA	785.917	615.773	469.045	400.385	339.106	273.926	207.546	133.243
BOOST TURBINE INLET	PSIA	785.917	615.773	469.045	400.385	339.106	273.926	207.546	133.243
DISCHARGE	PSIA	601.752	466.089	350.250	295.423	245.360	192.619	139.004	82.8640
1ST STAGE PUMP SUCTION	PSIA	601.752	466.089	350.250	295.423	245.360	192.619	139.004	82.8640
DISCHARGE	PSIA	3105.69	2343.43	1659.04	1351.28	1068.20	793.438	535.162	276.174
THRUST CHAMBER VALVE INLET	PSIA	3105.69	2343.43	1659.04	1351.28	1068.20	793.438	535.162	276.174
OX TC INJECTOR INLET	PSIA	2466.69	1893.45	1338.89	1134.23	911.240	688.630	471.722	250.617
2ND STAGE PUMP SUCTION	PSIA	2846.36	2224.99	1606.29	1319.14	1050.25	784.902	532.072	275.592
2ND STAGE PUMP DISCHARGE	PSIA	4665.91	3608.87	2560.17	2080.69	1637.80	1210.31	805.080	405.306
PREBURNER VALVE INLET	PSIA	4665.91	3608.87	2560.17	2080.69	1637.80	1210.31	805.080	405.306
DISCHARGE	PSIA	3847.98	2697.28	1830.99	1435.67	1130.80	813.307	546.021	275.986
INJECTOR INLET	PSIA	3847.98	2697.28	1830.99	1435.67	1130.80	813.307	546.021	275.986
SUMMARY HOT GAS PRESSURE SCHEDULE									
PREBURNER INJECTOR FACE	PSIA	3271.25	2409.02	1668.09	1338.02	1033.55	747.718	479.984	232.682
TURBINE INLET		3215.41	2372.30	1646.39	1322.16	1022.53	740.539	475.981	231.064
TURBINE EXIT / INJECTOR INLET	PSIA	1993.68	1575.88	1174.69	975.746	778.208	581.532	386.444	194.428
TC INJECTOR FACE	PSIA	1854.00	1483.20	1112.40	927.000	741.600	556.200	370.800	187.280

CASE 1 LETS2 O/S STAGED COMBUSTION CYCLE F=25K MRE=6.00 15 JUL 71 14:13
CASE 2 LETS2 O/S STAGED COMBUSTION CYCLE F=20K MRE=6.00 W/FPBYV, TTIT=1 15 JUL 71 14:21
CASE 3 LETS2 O/S STAGED COMBUSTION CYCLE F=15K MRE=6.00 15 JUL 71 14:56
CASE 4 LETS2 O/S STAGED COMBUSTION CYCLE F=12.5K MRE=6.00 15 JUL 71 14:56
CASE 5 LETS2 O/S STAGED COMBUSTION CYCLE F=10K MRE=6.00 19 JUL 71 14:42
CASE 6 LETS2 O/S STAGED COMBUSTION CYCLE F=7.5K MRE=6.00 19 JUL 71 14:43
CASE 7 LETS2 O/S STAGED COMBUSTION CYCLE F=5.0K MRE=6.00 19 JUL 71 14:43
CASE 8 LETS2 O/S STAGED COMBUSTION CYCLE F=2.5K MRE=6.0 16 JUL 71 08:54

TABLE LVIII

PERFORMANCE DATA, 10:1 THROTTLING MIXTURE RATIO AND THRUST EXCURSIONS

1811-01-001

ORBIT TO ORBIT SHUTTLE ENGINE   10:1 THROTTLEABLE   MIXTURE RATIO AND THRUST EXCURSIONS   PAGE 1

SUMMARY ENGINE PERFORMANCE DATA		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
ENGINE SPECIFIC IMPULSE	SEC	462.466	467.018	457.812	464.231	449.855	459.377	444.342	455.926
TOTAL OXIDIZER FLOW RATE	LB/SEC	46.9206	45.1885	23.6622	22.7450	9.62783	9.20765	4.87910	4.64948
TOTAL FUEL FLOW RATE	LB/SEC	7.21789	8.21718	3.64090	4.13498	1.48173	1.67400	.751087	.844936
TOTAL FLOW LEAVING SYSTEM	LB/SEC	54.1385	53.4057	27.3031	26.8800	11.1096	10.8816	5.63019	5.49441
OX TANK OUTFLOW	LB/SEC	46.9195	45.1909	23.6629	22.7442	9.62794	9.20838	4.87936	4.64948
FUEL TANK OUTFLOW	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
OXIDIZER PRESSURE-INTERFACE	PSIA	46.3390	46.3760	46.7200	46.7299	46.8285	46.8303	46.8445	46.8450
OXIDIZER ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
OXIDIZER TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
OXIDIZER DENSITY	LB/FT3	71.0865	71.0884	71.0877	71.0877	71.0874	71.0874	71.0874	71.0874
FUEL PRESSURE-INTERFACE	PSIA	20.6668	20.7357	21.1973	21.1645	21.2913	21.2862	21.3052	21.3039
FUEL ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.267	-106.260	-106.260
FUEL TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
FUEL DENSITY	LB/FT3	4.34944	4.34929	4.34980	4.34977	4.34991	4.34990	4.34992	4.34992
OXID. BOOST PUMP SPEED	RPM	13703.4	15459.5	9071.30	9975.84	5930.90	6343.26	4360.53	4647.78
OXID. MAIN PUMP SPEED	RPM	47429.6	52577.0	29957.3	32289.7	18035.6	18997.1	12501.7	13146.8
FUEL BOOST PUMP SPEED	RPM	22469.7	24469.9	15058.0	15416.0	9900.01	9983.41	7304.25	7381.63
FUEL MAIN PUMP SPEED	RPM	77266.5	83331.3	49045.1	51102.2	29662.9	30438.8	20633.4	21151.5
OXID PUMP DISCH PRESS-STG1	PSIA	2684.87	3599.86	1250.19	1482.30	508.697	561.293	266.291	286.572
OXID PUMP DISCH PRESS-STG2	PSIA	4101.00	5302.16	1934.63	2273.87	764.912	845.004	389.345	422.584
FUEL PUMP DISCH PRESS STG3	PSIA	4257.70	4756.37	1867.62	2000.99	716.357	755.756	360.452	379.670
PRIMARY COMBUSTOR PRESSURE	PSIA	3121.27	3472.97	1303.81	1382.59	471.520	490.330	226.591	234.699
THRUST CHAMBER PRESSURE	PSIA	1784.34	1818.25	892.170	909.126	356.868	363.650	178.434	181.825
OX. TC VALVE ADMITTANCE	KW	2.33486	1.15847	2.02024	1.03357	1.50723	.891456	.946161	.693716
OX. PB VALVE ADMITTANCE	KW	.250641	.178679	.808371-01	.751771-01	.381814-01	.377330-01	.226021-01	.226869-01
FU. CB BYP. VALVE ADMITTANCE	KW								
FU. PB BYP. VALVE ADMITTANCE	KW			.288031	.373881	.793816	.937255	1.13704	1.30954

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE — 10:1 THROTTLEABLE — MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
THRUST CHAMBER PERFORMANCE DATA									
THRUST CHAMBER PRESSURE	PSIA	1784.34	1818.25	892.170	909.126	356.868	363.650	178.434	181.825
TC INJECTOR MIXTURE RATIO	-	6.60768	5.58022	6.60509	5.58108	6.60194	5.57984	6.59882	5.58149
TC AREA RATIO	-	270.000	270.000	270.000	270.000	270.000	270.000	270.000	270.000
TC SPECIFIC IMPULSE	SEC	463.042	467.624	458.374	464.827	450.391	459.956	444.859	456.491
TC THRUST	LBS	25014.2	24918.1	12488.2	12466.8	4993.10	4994.11	2499.44	2502.71
DUMP COOLANT THRUST	LBS	22.9266	23.3623	11.4633	11.6812	4.58532	4.67246	2.29266	2.33623
DUMP COOLANT FLOW RATE	LB/SEC	.116972	.119196	.584862-01	.595978-01	.233945-01	.238391-01	.116972-01	.119196-01
REGEN COOLANT FLOW RATE	LB/SEC	6.89817	7.87145	3.48485	3.97135	1.42402	1.61419	.719064	.811905
INLET PRESSURE	PSIA	4205.73	4688.78	1854.22	1983.82	714.086	752.877	359.879	378.946
INLET TEMPERATURE	R	117.053	127.878	76.3117	77.7176	57.0592	56.8175	49.9840	49.8095
INLET DENSITY	LB/FT3	4.06950	4.05210	4.03861	4.07665	3.98452	4.02449	4.02622	4.05211
INLET ENTHALPY	BTU/LB	220.600	264.428	45.4188	52.1114	-35.8597	-35.8752	-64.3376	-64.5042
PRESSURE DROP	PSI	812.829	911.676	436.685	486.140	203.929	224.560	114.354	125.695
ENTHALPY RISE	BTU/LB	1350.63	1197.78	1539.26	1369.91	1832.59	1641.41	2109.64	1895.54
TEMPERATURE RISE	R	344.246	303.475	396.848	351.968	478.162	425.548	557.755	496.398
OUTLET PRESSURE	PSIA	3392.90	3777.10	1417.54	1497.68	510.157	528.317	245.525	253.251
OUTLET TEMPERATURE	R	461.299	431.352	473.160	429.685	535.221	482.365	607.739	546.208
OUTLET DENSITY	LB/FT3	1.10[illegible]70	1.38271	.528550	.609608	.175279	.201380	.760368-01	.865531-01
OUTLET ENTHALPY	BTU/LB	1571.23	1462.20	1584.68	1422.03	1796.75	1605.54	2045.32	1831.04
GAS INJECTOR FLOW RATE	LB/SEC	12.7713	14.5676	5.58034	6.32087	2.08117	2.33878	1.00551	1.12818
INLET PRESSURE	PSIA	1964.01	2031.38	963.214	990.911	381.936	391.783	190.294	195.057
PRESSURE DROP	PSI	126.144	158.584	44.2791	54.5110	14.3621	17.2228	6.50675	7.77726
AP/PCFACE	-	.686362-01	.846776-01	.481852-01	.582134-01	.390728-01	.459815-01	.354038-01	.415274-01
TC OXID. INJECTOR FLOW RATE	LB/SEC	41.1468	38.6191	21.6175	20.4488	8.98834	8.50173	4.60396	4.34433
INLET PRESSURE	PSIA	2406.65	2612.61	1147.94	1135.22	477.027	480.433	245.300	251.802
INLET TEMPERATURE	R	181.712	187.765	171.199	174.109	166.931	167.858	165.393	166.090
INLET ENTHALPY	BTU/LB	-45.4046	-41.5902	-51.5581	-50.2713	-54.4683	-54.0147	-55.4876	-55.2145
OUTLET TEMPERATURE	R	524.409	591.605	3[illegible]8.654	420.693	402.588	435.885	458.174	495.756
OUTLET DENSITY	LB/FT3	11.9906	10.0657	9.20855	8.06897	3.05005	2.88049	1.31896	1.23246
PRESSURE DROP	PSI	568.778	739.808	229.002	198.821	109.453	105.873	61.5128	64.5219
AP/PCFACE	-	.309477	.395028	.249204	.212324	.297770	.282660	.334696	.344521
TC IGNITER OXID. FLOW RATE	LB/SEC	.565463-01	.801888-01	.245788-01	.330856-01	.103173-01	.121357-01	.591898-02	.659990-02
FUEL FLOW RATE	LB/SEC	.996002-01	.103549	.477746-01	.491700-01	.171524-01	.179778-01	.953224-02	.981457-02
MIXTURE RATIO	-	.567733	.774400	.514474	.672881	.601506	.675038	.620943	.672460

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
PREBURNER PERFORMANCE DATA									
PREBURNER PRESSURE	PSIA	3121.27	3472.97	1303.81	1382.59	471.520	490.330	226.591	234.699
MIXTURE RATIO	-	.810123	.812044	.678535	.705182	.644222	.675642	.605647	.641127
GAS TEMPERATURE	R	1900.64	1866.02	1676.90	1676.85	1685.79	1685.35	1689.71	1689.57
SPECIFIC HT. RATIO	-	1.36396	1.36406	1.37170	1.37034	1.37363	1.37209	1.37555	1.37386
GAS CONSTANT	FT	421.293	422.248	456.111	448.922	465.718	456.906	477.018	466.605
FLOW RATE	LB/SEC	12.7161	14.4851	4.99849	5.46986	1.60630	1.72214	.714443	.764285
PB OXID. INJECTOR FLOW RATE	LB/SEC	5.64372	6.37857	1.97917	2.21067	.611497	.675152	.260463	.288554
INLET PRESSURE	PSIA	3641.72	4144.53	1396.78	1495.10	535.151	558.742	270.735	277.805
INLET TEMPERATURE	R	201.829	212.954	182.960	186.748	173.611	175.175	169.670	170.714
INLET DENSITY	LB/FT3	68.1440	66.6730	68.6811	67.9754	68.6349	69.2399	69.4435	69.2968
TEMPERATURE RISE	R	202.520	210.273	163.692	166.269	293.079	277.100	412.554	386.756
ENTHALPY RISE	BTU/LB	84.3271	83.7462	91.8490	90.4678	147.588	143.100	178.645	172.509
OUTLET TEMPERATURE	R	404.349	423.227	346.652	353.017	466.690	452.275	582.224	557.470
OUTLET DENSITY	LB/FT3	30.6096	30.5757	17.8316	18.5695	3.19319	3.48326	1.11416	1.24895
PRESSURE DROP	PSI	520.456	671.564	92.9687	112.506	63.6312	68.4123	44.1439	43.1063
ΔP/PCPB	-	.166745	.193369	.713055-01	.813736-01	.134949	.139523	.194817	.183666
PB FUEL INJECTOR FLOW RATE	LB/SEC	6.89711	7.87126	2.92907	3.15292	.959667	1.01076	.434738	.453499
INLET PRESSURE	PSIA	3374.15	3756.08	1406.76	1485.55	504.732	522.248	242.337	249.678
INLET TEMPERATURE	R	461.299	431.352	473.160	429.685	535.221	482.365	607.739	546.208
INLET DENSITY	LB/FT3	1.18381	1.37620	.524779	.604983	.173426	.199082	.750565-01	.853416-01
INLET ENTHALPY	BTU/LB	1571.23	1462.20	1584.68	1422.03	1796.75	1605.54	2045.32	1831.04
PRESSURE DROP	PSI	252.876	283.118	102.951	102.955	33.2122	31.9181	15.7458	14.9791
ΔP/PCPB	-	.810170-01	.815204-01	.789621-01	.744652-01	.704366-01	.650952-01	.694901-01	.638228-01
PB IGNITER OXID. FLOW RATE	LB/SEC	.708801-01	.112754	.415268-01	.513948-01	.176273-01	.197210-01	.900289-02	.980400-02
FUEL FLOW RATE	LB/SEC	.103547	.122307	.493286-01	.551023-01	.166996-01	.182208-01	.104742-01	.116567-01
MIXTURE RATIO	-	.684520	.921891	.841640	.932717	1.05555	1.08234	.859530	.841065

CASE 1	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
FUEL BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	19.5359	19.0113	20.8588	20.7278	21.2353	21.2146	21.2908	21.2857
ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
TEMPERATURE	R	37.3083	36.8992	37.7929	37.7941	37.7897	37.7899	37.7892	37.7893
SHAFT SPEED	RPM	22469.7	24469.9	15058.0	15416.0	9900.01	9983.41	7304.25	7381.63
WEIGHT FLOW RATE	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
VOLUME FLOW RATE	GPM	745.088	890.091	375.642	426.721	152.827	172.744	77.4502	87.2199
HEAD RISE (ACTUAL)	FT	879.600	970.323	495.701	485.351	259.922	255.140	153.015	153.488
PRESSURE RISE	PSI	27.7497	23.6449	15.5674	15.2394	8.13851	7.98820	4.78441	4.79923
(Q/N)/(QD/ND)	-	.957108	1.04991	.720043	.798954	.445569	.499429	.306054	.341047
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3797.41	4195.57	2778.05	3079.66	1890.61	2055.40	1477.53	1580.90
EFFICIENCY	-	.704591	.732642	.655789	.675357	.512326	.545908	.401706	.430303
NPSH (NO TSH)	FT	48.5949	24.3073	94.1311	89.6957	106.875	106.173	108.754	108.579
NPSP (NO TSH)	PSI	1.46728	.699327	2.84317	2.70911	3.22840	3.20718	3.28520	3.27993
THERMODYNAMIC SUCT. HEAD	FT	122.345	118.278	127.162	127.173	127.130	127.131	127.125	127.125
SUCTION SPECIFIC SPEED		12973.8	17692.7	5086.62	5635.04	2045.58	2198.05	1068.00	1146.00
S/SDESIGN	-	.288308	.393171	.113036	.125223	.454574-01	.488456-01	.237333-01	.254666-01
S/SMAX AT Q/QD	-	.290736	.496802	.131945	.134185	.979861-01	.945461-01	.585328-01	.575687-01
SHAFT HORSEPOWER	HP	16.3842	19.7856	5.00319	5.40341	1.36633	1.42271	.519890	.548251
DISCHARGE PRESSURE	PSIA	27.7497	23.6449	15.5674	15.2394	8.13851	7.98820	4.78441	4.79923
ENTHALPY	BTU/LB	-104.656	-104.558	-105.289	-105.336	-105.608	-105.659	-105.771	-105.802
TEMPERATURE	R	38.0966	38.1955	37.9802	37.9669	37.9436	37.9252	37.9217	37.9091
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
INLET PRESSURE	PSIA	191.058	199.606	109.571	110.755	60.9780	61.3059	42.6311	43.1069
ENTHALPY	BTU/LB	-96.3780	-95.5630	-100.984	-101.033	-103.314	-103.441	-104.290	-104.363
TEMPERATURE	R	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337	39.5337
PRESSURE DROP	PSI	37.8809	40.1662	22.8744	21.7581	15.3033	14.0948	11.4739	10.7412
ISEN. SPOUTING VELOCITY	FT/SEC	324.173	334.961	254.107	246.584	218.895	206.816	202.148	191.756
MEAN WHEEL SPEED	FT/SEC	199.674	217.449	133.811	136.993	87.9751	88.7162	64.9083	65.5958
VELOCITY RATIO - U/CO	-	.615948	.649174	.526591	.555560	.401905	.428962	.321091	.342080
EFFICIENCY	-	.764460	.759862	.753549	.760803	.681612	.702835	.599696	.623639
SHAFT HORSEPOWER	HP	16.3851	19.7931	5.00496	5.40522	1.36688	1.42214	.519785	.547738
OUTLET PRESSURE	PSIA	153.177	159.440	86.6963	88.9964	45.6748	47.2111	31.1572	32.3658
ENTHALPY	BTU/LB	-97.9824	-97.2656	-101.956	-101.956	-103.966	-104.042	-104.780	-104.821
TEMPERATURE	R	39.6719	39.9205	38.7232	38.6954	38.3974	38.3474	38.2532	38.2209
TURBINE TORQUE	FT-LBS	3.82969	4.24830	1.74569	1.84151	.725150	.748166	.373750	.389722

CASE 1	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
FUEL PUMP - HIGH SPEED INDUCER AND FIRST STAGE									
HIGH SPEED FUEL INDUCER									
SUCTION PRESSURE	PSIA	47.2856	42.6562	36.4262	35.9672	29.3738	29.2028	26.0752	26.0849
ENTHALPY	BTU/LB	-104.656	-104.558	-105.289	-105.336	-105.608	-105.659	-105.771	-105.802
TEMPERATURE	R	38.0868	38.1828	37.9779	37.9639	37.9432	37.9248	37.9216	37.9090
SHAFT SPEED	RPM	77266.5	83331.3	49045.1	51102.2	29662.9	30438.8	633.4	21151.5
WEIGHT FLOW RATE	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
VOLUME FLOW RATE	GPM	744.702	849.032	375.650	426.682	152.920	172.819	77.5122	87.2782
HEAD RISE (ACTUAL)	FT	4554.32	5031.05	2236.46	2236.72	989.346	1005.36	523.117	537.916
PRESSURE RISE	PSI	143.772	156.950	73.1445	74.7873	31.6042	32.1031	16.5559	17.0220
(Q/N)/(QD/ND)	-	.973545	1.02916	.773666	.843392	.520735	.573496	.379458	.416802
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	3803.34	4041.07	2922.92	3192.14	2079.38	2241.20	1660.76	1769.12
EFFICIENCY	-	.707016	.718739	.667657	.682757	.554191	.582461	.454142	.480616
NPSH (NO TSH)	FT	942.031	780.958	592.804	578.848	362.669	358.630	255.421	256.856
NPSP (NO TSH)	PSI	28.4585	23.5549	17.9049	17.4847	10.9485	10.8284	7.70950	7.75384
THERMODYNAMIC SUCT. HEAD	FT	130.083	131.037	129.000	128.861	128.656	128.472	128.441	128.316
SUCTION SPECIFIC SPEED		11253.9	14631.1	6826.10	7693.08	3514.95	3859.30	2094.71	2272.76
S/SDESIGN	-	.375129	.487702	.227537	.256436	.117165	.128643	.698238-01	.757586-01
S/SMAX AT Q/QD	-	.378492	.561208	.249417	.256576	.213605	.202020	.147594	.162206
SHAFT HORSEPOWER	HP	84.5418	104.571	22.1717	25.1821	4.80780	5.25428	1.57215	1.72027
DISCHARGE PRESSURE	PSIA	191.058	199.606	109.571	110.755	60.9780	61.3059	42.6311	43.1069
ENTHALPY	BTU/LB	-96.3780	-95.5630	-100.984	-101.033	-103.314	-103.441	-104.290	-104.363
TEMPERATURE	R	39.9871	40.2687	38.8630	38.8286	38.4729	38.4165	36.3043	38.2687
FUEL PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
VOLUME FLOW RATE (1)	GPM	865.060	992.634	426.678	481.088	177.642	198.313	93.3088	103.518
HEAD RISE (ACTUAL)	FT	46893.9	52898.4	20305.9	21781.0	7668.41	8078.04	3749.25	3937.22
PRESSURE RISE	PSI	1385.96	1565.82	605.662	651.817	224.405	236.753	107.958	113.943
(Q/N)/(QD/ND)		.978867	1.04148	.760632	.823102	.523604	.569630	.395385	.427903
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	713.144	752.697	595.555	625.163	480.861	503.064	415.981	432.970
EFFICIENCY	-	.605561	.601119	.563619	.579356	.462481	.486466	.383113	.402950
NPSH (NO TSH)	FT	3416.16	3464.54	1727.53	1807.78	631.312	698.394	255.339	304.193
NPSP (NO TSH)	PSI	95.2145	95.4521	50.0843	52.4425	18.6075	20.6148	7.58132	9.04007
THERMODYNAMIC SUCT. HEAD	FT	215.323	224.266	175.564	175.432	153.987	152.814	144.836	144.135
SUCTION SPECIFIC SPEED		4857.93	5546.75	3516.01	3771.59	2665.08	2720.05	2227.64	2208.78
S/SDESIGN	-	.485793	.554675	.351601	.377159	.266508	.272005	.222764	.220878
S/SMAX AT Q/QD	-	.476591	.587869	.404494	.402625	.461634	.439936	.465855	.422569
SHAFT HORSEPOWER	HP	1089.22	1403.97	259.992	306.073	50.6465	56.7062	15.8167	17.5358
DISCHARGE PRESSURE	PSIA	1539.14	1725.26	692.358	740.813	270.080	283.964	139.115	146.309
ENTHALPY	BTU/LB	21.3181	37.7028	-45.7946	-43.7544	-77.5031	-77.7198	-88.9244	-89.0837
TEMPERATURE	R	70.7292	74.7842	54.0023	54.3050	46.3539	46.2074	43.2323	43.1216

(1) VOLUME FLOW INCLUDES 7% RECIRCULATION

CASE 1 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2 LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4 LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6 LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7 LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8 LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 10:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
FUEL PUMP - SECOND AND THIRD STAGES									
SECOND STAGE FUEL PUMP									
SUCTION PRESSURE	PSIA	1539.14	1725.26	692.358	740.813	270.080	283.964	139.115	146.309
ENTHALPY	BTU/LB	21.3181	37.7028	-45.7946	-43.7544	-77.5031	-77.7198	-88.9244	-89.0837
TEMPERATURE	R	70.7100	74.7559	53.9983	54.2993	46.3527	46.2060	43.2322	43.1214
SHAFT SPEED	RPM	77266.5	83331.3	49045.1	51102.2	29662.9	30438.8	20633.4	21151.5
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
VOLUME FLOW RATE (1)	GPM	859.465	981.236	433.906	487.378	181.947	202.431	95.3742	105.620
HEAD RISE (ACTUAL)	FT	47028.7	53174.0	20254.6	21720.0	7694.18	8067.92	3749.26	3935.91
PRESSURE RISE	PSI	1362.67	1542.38	591.609	634.352	226.295	238.609	114.173	120.066
(G/N)/(QU/ND)	-	.973600	1.03065	.774363	.834777	.536877	.582096	.404579	.437068
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	709.305	745.453	601.728	630.562	487.039	508.739	420.560	437.452
EFFICIENCY	-	.605497	.602704	.567668	.580500	.470817	.491790	.388407	.408192
SHAFT HORSEPOWER	HP	1092.47	1407.58	257.486	304.614	49.9169	56.0221	15.6011	17.3049
DISCHARGE PRESSURE	PSIA	2901.81	3267.65	1283.97	1375.17	496.375	522.573	253.288	266.374
ENTHALPY	BTU/LB	121.127	151.077	.562329-01	4.32678	-56.5026	-56.6383	-76.5200	-76.6929
TEMPERATURE	R	94.2464	101.655	65.2860	66.2052	51.7851	51.6189	46.7889	46.6671
THIRD STAGE FUEL PUMP									
WT. FLOW RATE (DELIVERED)	LB/SEC	7.21838	8.21652	3.64044	4.13531	1.48122	1.67425	.750671	.845361
VOLUME FLOW RATE (1)	GPM	856.200	972.386	438.069	491.005	185.632	206.006	97.1495	107.389
HEAD RISE (ACTUAL)	FT	46876.3	53152.6	20197.9	21636.1	7683.31	8053.96	3748.09	3934.02
PRESSURE RISE	PSI	1355.89	1488.72	583.656	625.826	219.982	233.182	107.164	113.295
(G/N)/(QU/ND)	-	.979547	1.03151	.789566	.849353	.553199	.598266	.416208	.448806
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	709.682	742.308	605.882	634.744	492.468	513.879	424.554	441.259
EFFICIENCY	-	.605567	.602579	.572170	.581846	.478294	.498463	.395363	.414758
SHAFT HORSEPOWER	HP	1086.80	1407.30	254.740	302.735	49.0671	55.1764	15.3219	17.0227
FUEL TURBINE SPEED	RPM	77266.5	83331.3	49045.1	51102.2	29662.9	30438.8	20633.4	21151.5
COMBINED FUEL PUMP PERFORMANCE BASED ON DELIVERED FLOW									
TOTAL FUEL TPA POWER REQ'D	HP	3355.03	4323.42	794.393	938.604	154.438	173.159	48.3118	53.5837
TOTAL FUEL TPA HEAD RISE	FT	145353.	164256.	62994.8	67423.8	24035.3	25205.3	11769.7	12345.1
OVERALL PUMP EFFICIENCY	-	.568598	.567570	.524881	.540103	.419132	.443103	.332506	.354112

(1) VOL. FLOW INCLUDES 7% RECIRC.

CASE 1	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 3	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71 14:57
CASE 4	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:58
CASE 5	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 15:01
CASE 6	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 15:35
CASE 7	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71 08:55
CASE 8	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71 08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49893	5.50064	6.49770	5.50038	6.49605	5.50275
OXIDIZER BOOST PUMP AND HYDRAULIC TURBINE									
LOW SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	44.8143	44.9615	46.3322	46.3716	46.7643	46.7716	46.8280	46.8300
ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
TEMPERATURE	R	162.928	162.927	162.915	162.914	162.904	162.904	162.903	162.903
SHAFT SPEED	RPM	13703.4	15459.5	9071.30	9975.84	5930.90	6343.26	4360.53	4647.78
WEIGHT FLOW RATE	LB/SEC	46.9195	45.1909	23.6629	22.7442	9.62794	9.20838	4.87936	4.64948
VOLUME FLOW RATE	GPM	296.201	285.290	149.390	143.590	60.7845	58.1356	30.8051	29.3538
HEAD RISE (ACTUAL)	FT	208.407	309.205	116.342	153.675	62.7827	74.3903	37.3415	43.2439
PRESSURE RISE	PSI	102.865	152.645	57.4341	75.8640	30.9935	36.7237	18.4341	21.3479
(Q/N)/(QD/ND)	-	1.06510	.909338	.811495	.709267	.505017	.451610	.348111	.311210
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4299.70	3541.23	3129.87	2738.79	2073.18	1909.40	1602.17	1493.26
EFFICIENCY	-	.671241	.655669	.625903	.606923	.505402	.477277	.399743	.374205
NPSH (NO TSH)	FT	60.0662	60.3663	63.1627	63.2441	64.0550	64.0700	64.1865	64.1907
NPSP (NO TSH)	PSI	29.6542	29.8023	31.1815	31.2217	31.6216	31.6291	31.6865	31.6886
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		10414.8	11490.7	4725.89	5090.63	1951.52	2040.88	1019.95	1061.17
S/SDESIGN	-	.231440	.255348	.105020	.113125	.433670-01	.453528-01	.226655-01	.235815-01
S/SMAX AT Q/QD	-	.313594	.258840	.111444	.134081	.826643-01	.971591-01	.505109-01	.573041-01
SHAFT HORSEPOWER	HP	26.4865	38.7481	7.99716	10.4708	2.17457	2.60956	.828725	.976916
DISCHARGE PRESSURE	PSIA	102.885	152.645	57.4341	75.8640	30.9935	36.7237	18.4341	21.3479
ENTHALPY	BTU/LB	-56.6460	-56.4390	-56.8061	-56.7196	-56.8854	-56.8447	-56.9250	-56.8965
TEMPERATURE	R	163.046	163.222	163.050	163.056	162.999	163.025	163.021	163.068
HYDRAULIC TURBINE									
WEIGHT FLOW RATE	LB/SEC	46.9195	45.1909	23.6629	22.7442	9.62794	9.20838	4.87936	4.64948
INLET PRESSURE	PSIA	666.861	933.671	364.432	449.685	195.914	219.343	128.225	139.444
ENTHALPY	BTU/LB	-54.5909	-53.5026	-55.7306	-55.3374	-56.3129	-56.1566	-56.5558	-56.4553
TEMPERATURE	R	165.309	165.309	165.309	165.309	165.309	165.309	165.309	165.309
PRESSURE DROP	PSI	152.485	232.160	91.2533	124.483	61.0403	76.5706	45.9318	56.8299
ISEN. SPOUTING VELOCITY	FT/SEC	161.777	199.355	125.784	148.030	107.908	122.839	99.7993	113.339
MEAN WHEEL SPEED	FT/SEC	101.606	114.626	67.2602	73.9670	43.9753	47.0328	32.3317	34.4615
VELOCITY RATIO - U/CO	-	.626057	.574984	.534726	.499673	.407526	.382881	.323966	.304057
EFFICIENCY	-	.763327	.763670	.[illegible]35946	.743611	.686277	.664827	.603086	.578887
SHAFT HORSEPOWER	HP	26.4851	38.7537	7.[illegible]9671	10.4718	2.17390	2.61015	.828131	.976917
OUTLET PRESSURE	PSIA	514.376	701.511	273.179	325.202	134.874	142.772	82.2928	82.6145
ENTHALPY	BTU/LB	-54.9899	-54.1087	-55.9695	-55.6628	-56.4725	-56.3570	-56.6758	-56.6038
TEMPERATURE	R	165.591	167.140	164.213	164.743	163.592	163.791	163.454	163.626
TURBINE TORQUE	FT-LBS	10.1509	13.1659	4.62993	5.51322	1.92510	2.16116	.997456	1.10394

CASE 1	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 O2S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 O2S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 O2S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 O2S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE   10:1 THROTTLEABLE   MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
OXIDIZER PUMP - HIGH SPEED INDUCER AND MAIN STAGE									
HIGH SPEED OXIDIZER INDUCER									
SUCTION PRESSURE	PSIA	147.699	197.607	103.766	122.236	77.7578	83.4953	65.2621	68.1779
ENTHALPY	BTU/LB	-56.6460	-56.4390	-56.8061	-56.7196	-56.8854	-56.8447	-56.9250	-56.8965
TEMPERATURE	R	163.070	163.223	163.060	163.051	163.001	163.027	163.021	163.068
SHAFT SPEED	RPM	47429.6	52577.0	29957.3	32289.7	18036.6	18997.1	12501.7	13146.8
WEIGHT FLOW RATE	LB/SEC	46.9195	45.1909	23.6629	22.7442	9.62794	9.20838	4.87936	4.64948
VOLUME FLOW RATE	GPM	297.030	286.310	149.587	143.946	60.8731	58.2423	30.8390	29.3914
HEAD RISE (ACTUAL)	FT	1054.53	1496.27	528.712	664.941	239.694	275.687	127.682	144.548
PRESSURE RISE	PSI	519.162	736.065	260.666	327.449	118.156	135.848	62.9625	71.2666
(G/N)/(GD/ND)	-	1.07975	.938885	.860920	.768614	.581892	.528597	.425309	.385455
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	4400.60	3697.90	3323.03	2958.54	2310.07	2142.86	1827.77	1709.70
EFFICIENCY	-	.659387	.654806	.631705	.618183	.538059	.514871	.444484	.421031
NPSH (NO TSH)	FT	269.002	370.433	179.523	217.250	126.878	138.530	101.463	107.324
NPSP (NO TSH)	PSI	132.434	182.228	88.5084	106.984	62.5438	68.2622	50.0333	52.9142
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		12171.0	10451.6	7348.20	6753.10	3636.79	3514.59	2109.57	2079.64
S/SDESIGN	-	.405699	.348387	.244940	.225103	.121226	.117153	.703190-01	.693212-01
S/SMAX AT Q/QD	-	.588682	.351593	.252373	.247997	.186174	.208796	.151946	.145302
SHAFT HORSEPOWER	HP	136.430	187.753	36.0088	44.4810	7.79827	8.96477	2.54843	2.90228
DISCHARGE PRESSURE	PSIA	666.861	933.671	364.432	449.685	195.914	219.343	128.225	139.444
ENTHALPY	BTU/LB	-54.5909	-53.5026	-55.7306	-55.3374	-56.3129	-56.1566	-56.5558	-56.4553
TEMPERATURE	R	166.057	167.810	164.273	165.201	163.535	163.756	163.432	163.587
OXIDIZER PUMP - STAGE 1									
WT. FLOW RATE (DELIVERED)	LB/SEC	46.9195	45.1909	23.6629	22.7442	9.62794	9.20838	4.87936	4.64948
VOLUME FLOW RATE (1)	GPM	312.739	303.724	159.058	154.129	66.3103	63.9930	34.5481	33.2979
HEAD RISE (ACTUAL)	FT	4424.90	5912.59	1988.74	2358.01	760.030	851.618	373.780	414.512
PRESSURE RISE	PSI	2170.50	2898.35	977.009	1157.10	373.823	418.521	183.998	203.958
(G/N)/(GD/ND)		1.08450	.950124	.873270	.785085	.604676	.554042	.454520	.416575
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	1546.01	1358.95	1268.66	1184.67	1014.66	963.982	864.406	825.800
EFFICIENCY	-	.620123	.636649	.607630	.591172	.514893	.494712	.428988	.407695
NPSH (NO TSH)	FT	1014.34	1395.72	523.116	628.946	242.149	258.082	135.380	135.790
NPSP (NO TSH)	PSI	496.347	681.872	256.675	308.151	119.022	126.728	66.6148	66.7806
THERMODYNAMIC SUCT. HEAD	FT	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000	4.00000
SUCTION SPECIFIC SPEED		4652.66	4004.09	3434.39	3176.74	2363.45	2333.05	1811.46	1866.04
S/SDESIGN	-	.465286	.400409	.343439	.317674	.236345	.233305	.181146	.186604
S/SMAX AT Q/QD	-	.537138	.388476	.350121	.354055	.357283	.387707	.340879	.378527
SHAFT HORSEPOWER	HP	637.024	803.911	149.014	175.717	28.0662	31.5564	8.64089	9.71289
DISCHARGE PRESSURE	PSIA	2684.87	3599.86	1250.19	1482.30	508.697	561.293	266.291	286.572
ENTHALPY	BTU/LB	-45.4046	-41.5902	-51.5581	-50.2713	-54.4683	-54.0147	-55.4876	-55.2145
TEMPERATURE	R	181.712	187.765	171.199	174.109	166.931	167.858	165.393	166.090

(1) VOLUME FLOW INCLUDES 5% RECIRCULATION

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12493.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
OXIDIZER PUMP - HALF STAGE									
SUCTION PRESSURE	PSIA	2452.64	3300.23	1221.18	1445.96	505.879	557.857	265.775	285.938
ENTHALPY	BTU/LB	-45.4046	-41.5902	-51.5581	-50.2713	-54.4683	-54.0147	-55.4876	-55.2145
TEMPERATURE	R	183.799	190.474	172.016	175.082	167.473	168.424	165.755	166.479
SHAFT SPEED	RPM	47429.6	52577.0	29957.3	32289.7	18036.6	18997.1	12501.7	13146.8
WEIGHT FLOW RATE	LB/SEC	5.71434	6.49130	2.02060	2.26207	.629362	.694390	.269486	.298578
VOLUME FLOW RATE	GPM	38.6798	43.9716	14.1625	15.7990	4.77545	5.23969	2.24209	2.45699
HEAD RISE (ACTUAL)	FT	3366.26	4091.64	1457.06	1690.02	529.683	588.063	252.313	279.283
PRESSURE RISE	PSI	1646.16	2001.93	713.454	827.907	259.033	287.147	123.570	136.645
(1) (Q/N)/(QD/ND)	-	.926729	.950373	.537223	.556009	.300869	.313427	.203799	.212374
SPECIFIC SPEED	RPM-GPM**.5/FT**.75	667.468	681.488	478.039	486.924	356.985	364.142	295.692	301.639
EFFICIENCY	-	.452608	.455532	.361470	.370706	.244358	.251942	.181675	.187960
SHAFT HORSEPOWER	HP	81.5064	111.677	16.2580	20.5166	2.94960	3.47898	.888835	1.04104
DISCHARGE PRESSURE	PSIA	4101.00	5302.16	1934.63	2273.87	764.912	845.004	389.345	422.584
ENTHALPY	BTU/LB	-35.3751	-29.4811	-46.0550	-44.0502	-51.4737	-50.7923	-53.5549	-53.1479
TEMPERATURE	R	201.829	212.954	182.960	186.748	173.611	175.175	169.670	170.714

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
GAS TURBINE PERFORMANCE DATA									
FULL TPA SHAFT SPEED	RPM	77266.5	83331.3	49045.1	51102.2	29662.9	30438.8	20633.4	21151.5
TURBINE INLET TEMPERATURE	R	1882.40	1847.88	1660.08	1659.72	1660.24	1659.46	1660.70	1659.99
WEIGHT FLOW RATE (1)	LB/SEC	9.45335	10.7693	3.71630	4.06673	1.19440	1.28006	.531365	.568166
MEAN WHEEL SPEED	FT/SEC	1254.56	1353.04	796.337	829.738	481.631	494.230	335.021	343.433
ISEN. SPOUTING VELOCITY	FT/SEC	4636.24	4929.32	3694.19	3837.07	3131.37	3197.30	2891.42	2941.45
STAGE VEL. RATIO U/C	-	.382684	.388183	.304854	.305812	.217517	.218605	.163861	.165118
INLET PRESSURE (TOTAL)	PSIA	3070.41	3410.63	1288.93	1365.40	467.693	486.103	225.058	233.019
EXIT PRESSURE (STATIC)	PSIA	1964.01	2031.38	963.2[illegible]	990.911	381.936	391.783	190.294	195.057
PRESSURE RATIO	-	1.56334	1.67897	1.33816	1.37793	1.22453	1.24075	1.18269	1.19462
SHAFT HORSEPOWER	HP	3355.22	4323.89	794.267	938.794	154.378	173.024	48.2397	53.6830
SHAFT TORQUE	FT-LBS	228.068	272.522	85.0561	96.4862	27.3343	29.8547	12.2791	13.3300
EXIT TEMPERATURE	R	1758.75	1708.30	1590.25	1583.28	1618.72	1615.34	1632.13	1629.68
EFFICIENCY	-	.628218	.628664	.595833	.596528	.501504	.503055	.413139	.415474
OXID TPA SHAFT SPEED	RPM	47429.6	52577.0	29957.3	32289.7	18036.6	18997.1	12501.7	13146.8
TURBINE INLET TEMPERATURE	R	1882.40	1847.88	1660.08	1659.72	1660.24	1659.46	1660.70	1659.99
WEIGHT FLOW RATE (1)	LB/SEC	3.26236	3.71651	1.28250	1.40344	.412188	.441750	.183375	.196075
MEAN WHEEL SPEED	FT/SEC	1042.62	1155.77	658.535	709.808	396.489	417.602	274.817	288.999
ISEN. SPOUTING VELOCITY	FT/SEC	4636.24	4929.32	3694.19	3837.07	3131.37	3197.30	2891.42	2941.45
VELOCITY RATIO - U/C	-	.224884	.234468	.178262	.184987	.126618	.130611	.950457-01	.982500-01
INLET PRESSURE (TOTAL)	PSIA	3070.41	3410.63	1288.93	1365.40	467.693	486.103	225.058	233.019
EXIT PRESSURE (STATIC)	PSIA	1964.01	2031.38	963.214	990.911	381.936	391.783	190.294	195.057
PRESSURE RATIO	-	1.56334	1.67897	1.33816	1.37793	1.22453	1.24075	1.18269	1.19462
SHAFT HORSEPOWER	HP	854.774	1103.14	201.040	240.657	38.7757	44.1742	12.0503	13.6670
SHAFT TORQUE	FT-LBS	94.6534	110.197	35.2463	39.1442	11.2912	12.2128	5.06248	5.45994
EXIT TEMPERATURE	R	1791.12	1744.70	1608.86	1602.94	1630.02	1626.82	1640.02	1637.63
EFFICIENCY	-	.463760	.464761	.437012	.443110	.365007	.372162	.299049	.306502
TURBINE GAS PROP RTIES									
SPECIFIC HEAT RATIO	-	1.36396	1.36406	1.37170	1.37034	1.37363	1.37209	1.37555	1.37386
GAS CONSTANT	FT	421.293	422.248	456.111	448.922	465.718	456.906	477.018	466.605
MOLECULAR WEIGHT	-	3.66491	3.65662	3.38514	3.43935	3.31531	3.37925	3.23678	3.30901
TEMPERATURE	R	1882.40	1847.88	1660.08	1659.72	1660.24	1659.46	1660.70	1659.99

(1) INCLUDES 7% LEAKAGE FLOW

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
CONTROLS COMPONENTS									
OXID. THRUST CHAMBER VALVE									
WEIGHT FLOW RATE	LB/SEC	41.1466	38.6194	21.6177	20.4491	8.98826	8.50185	4.60395	4.34431
INLET PRESSURE	PSIA	2084.67	3599.86	1250.19	1482.30	508.697	561.293	266.291	286.572
INLET TEMPERATURE	R	181.712	187.765	171.199	174.109	166.931	167.858	165.393	166.090
INLET DENSITY	LB/FT3	70.5042	70.4553	70.5098	70.5428	70.4209	70.3141	70.5238	70.4552
PRESSURE DROP	PSI	274.869	984.272	101.332	346.256	31.5119	80.7179	20.9498	34.7335
9 ADMITTANCE	KW	2.33486	1.15847	2.02024	1.03357	1.50723	.891456	.946161	.693716
OXID. PREBURNER VALVE									
WEIGHT FLOW RATE	LB/SEC	5.64346	6.37854	1.97907	2.21068	.611735	.674669	.260483	.288774
INLET PRESSURE	PSIA	4101.00	5302.16	1934.63	2273.87	764.912	845.004	389.345	422.584
INLET TEMPERATURE	R	201.829	212.954	182.960	186.748	173.611	175.175	169.670	170.714
INLET DENSITY	LB/FT3	68.6806	68.6928	69.5380	69.2878	69.7158	69.6884	69.8758	69.8305
PRESSURE DROP	PSI	459.277	1157.63	537.855	778.767	229.761	286.262	118.610	144.779
8 ADMITTANCE	KW	.250641	.178679	.808371-01	.751771-01	.381814-01	.377330-01	.226021-01	.226869-01
FUEL PREBURNER BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC			.556294	.818759	.463741	.604685	.284485	.357919
INLET PRESSURE	PSIA			1406.76	1485.55	504.732	522.248	242.337	249.678
INLET TEMPERATURE	R			473.160	429.685	535.221	482.365	607.739	546.208
INLET DENSITY	LB/FT3			.524779	.604983	.173426	.199082	.750565-01	.853416-01
PRESSURE DROP	PSI			443.545	494.637	122.796	130.465	52.0432	54.6209
ADMITTANCE	KW			.288031	.373881	.793816	.937255	1.13704	1.30954
FUEL COOL. JKT. BYPASS CIRCUIT									
WEIGHT FLOW RATE	LB/SEC								
INLET PRESSURE	PSIA	4205.73	4688.78	1854.22	1983.82	714.086	752.877	359.879	378.946
INLET TEMPERATURE	R	117.053	127.878	76.3117	77.7176	57.0592	56.8175	49.9840	49.8095
INLET DENSITY	LB/FT3	4.06950	4.05210	4.03861	4.07665	3.98452	4.02449	4.02622	4.05211
PRESSURE DROP	PSI	2241.72	2657.39	891.006	992.912	332.150	361.094	169.586	183.889
ADMITTANCE	KW								
FUEL SHUTOFF VALVE									
WEIGHT FLOW RATE	LB/SEC	7.01524	7.99067	3.54334	4.03104	1.44737	1.63805	.730664	.823889
INLET PRESSURE	PSIA	4257.70	4756.37	1867.62	2000.99	716.357	755.756	360.452	379.670
INLET TEMPERATURE	R	117.053	127.878	76.3117	77.7176	57.0592	56.8175	49.9840	49.8095
INLET DENSITY	LB/FT3	4.09248	4.08212	4.04889	4.08980	3.98685	4.02744	4.02691	4.05299
PRESSURE DROP	PSI	51.9656	67.5923	13.4001	17.1692	2.27063	2.87901	.572903	.723735
ADMITTANCE	KW	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000	3.80000

CASE 1	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71	14:14
CASE 2	LETS2 OOS STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71	14:15
CASE 3	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=6.50	15 JUL 71	14:57
CASE 4	LETS2 OOS STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71	14:58
CASE 5	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71	15:01
CASE 6	LETS2 OOS STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71	15:35
CASE 7	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71	08:55
CASE 8	LETS2 OOS STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71	08:55

TABLE LVIII (cont.)

ORBIT TO ORBIT SHUTTLE ENGINE 10:1 THROTTLEABLE MIXTURE RATIO AND THRUST EXCURSIONS

		CASE 1	CASE 2	CASE 3	CASE 4	CASE 5	CASE 6	CASE 7	CASE 8
ENGINE THRUST	LBS	25037.2	24941.4	12499.7	12478.5	4997.69	4998.78	2501.73	2505.05
ENGINE MIXTURE RATIO	-	6.50059	5.49928	6.49898	5.50064	6.49770	5.50038	6.49605	5.50275
SUMMARY FUEL PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100	21.3100
TANK OUTLET TEMPERATURE	R	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891	37.7891
TANK OUTLET ENTHALPY	BTU/LB	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260	-106.260
BOOST PUMP SUCTION	PSIA	19.5359	19.0113	20.8588	20.7278	21.2353	21.2146	21.2908	21.2857
DISCHARGE	PSIA	47.2856	42.6562	36.4262	35.9672	29.3738	29.2028	26.0752	26.0849
HIGH SPEED INDUCER SUCTION	PSIA	47.2856	42.6562	36.4262	35.9672	29.3738	29.2028	26.0752	26.0849
DISCHARGE	PSIA	191.058	199.606	109.571	110.755	60.9780	61.3059	42.6311	43.1069
BOOST TURBINE INLET	PSIA	191.058	199.606	109.571	110.755	60.9780	61.3059	42.6311	43.1069
DISCHARGE	PSIA	153.177	159.440	86.6963	88.9964	45.6748	47.2111	31.1572	32.3658
1ST STAGE PUMP SUCTION	PSIA	153.177	159.440	86.6963	88.9964	45.6748	47.2111	31.1572	32.3658
DISCHARGE	PSIA	1539.14	1725.26	692.358	740.813	270.080	283.964	139.115	146.309
2ND STAGE PUMP DISCHARGE	PSIA	2901.81	3267.65	1283.97	1375.17	496.375	522.573	253.288	266.374
3RD STAGE PUMP DISCHARGE	PSIA	4257.70	4756.37	1867.62	2000.99	716.357	755.756	360.452	379.670
COOLING JACKET INLET	PSIA	4205.73	4688.78	1854.22	1983.82	714.086	752.877	359.879	378.946
DISCHARGE	PSIA	3392.90	3777.10	1417.54	1497.68	510.157	528.317	245.525	253.251
PREBURNER INJECTOR INLET	PSIA	3374.15	3756.08	1406.76	1485.55	504.732	522.248	242.337	249.678
SUMMARY OXIDIZER PRESSURE SCHEDULE									
TANK OUTLET PRESSURE	PSIA	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500	46.8500
TANK OUTLET TEMPERATURE	R	162.902	162.902	162.902	162.902	162.902	162.902	162.902	162.902
TANK OUTLET ENTHALPY	BTU/LB	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450	-57.0450
BOOST PUMP SUCTION	PSIA	44.8143	44.9615	46.3322	46.3716	46.7643	46.7716	46.8280	46.8300
DISCHARGE	PSIA	147.699	197.607	103.766	122.236	77.7578	83.4953	65.2621	68.1779
HIGH SPEED INDUCER SUCTION	PSIA	147.699	197.607	103.766	122.236	77.7578	83.4953	65.2621	68.1779
DISCHARGE	PSIA	666.861	933.671	364.432	449.685	195.914	219.343	128.225	139.444
BOOST TURBINE INLET	PSIA	666.861	933.671	364.432	449.685	195.914	219.343	128.225	139.444
DISCHARGE	PSIA	514.376	701.511	273.179	325.202	134.874	142.772	82.2928	82.6145
1ST STAGE PUMP SUCTION	PSIA	514.376	701.511	273.179	325.202	134.874	142.772	82.2928	82.6145
DISCHARGE	PSIA	2684.87	3599.86	1250.19	1482.30	508.697	561.293	266.291	286.572
THRUST CHAMBER VALVE INLET	PSIA	2684.87	3599.86	1250.19	1482.30	508.697	561.293	266.291	286.572
OX TC INJECTOR INLET	PSIA	2400.65	2612.61	1147.94	1135.22	477.027	480.433	245.300	251.802
2ND STAGE PUMP SUCTION	PSIA	2452.84	3300.23	1221.18	1445.96	505.879	557.857	265.775	285.938
2ND STAGE PUMP DISCHARGE	PSIA	4101.00	5302.16	1934.63	2273.87	764.912	845.004	389.345	422.584
PREBURNER VALVE INLET	PSIA	4101.00	5302.16	1934.63	2273.87	764.912	845.004	389.345	422.584
DISCHARGE	PSIA	3641.72	4144.53	1396.78	1495.10	535.151	558.742	270.735	277.805
INJECTOR INLET	PSIA	3641.72	4144.53	1396.78	1495.10	535.151	558.742	270.735	277.805
SUMMARY HOT GAS PRESSURE SCHEDULE									
PREBURNER INJECTOR FACE	PSIA	3121.27	3472.97	1303.61	1382.59	471.520	490.330	226.591	234.699
TURBINE INLET		3070.41	3410.63	1288.93	1365.40	467.693	486.103	225.058	233.019
TURBINE EXIT / INJECTOR INLET	PSIA	1964.01	2031.38	963.214	990.911	381.936	391.783	190.294	195.057
TC INJECTION FACE	PSIA	1837.87	1872.80	918.935	936.400	367.574	374.560	183.787	187.280

CASE 1	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=6.50	15 JUL 71 14:14
CASE 2	LETS2 O/S STAGED COMBUSTION CYCLE	F=25K	MRE=5.50	15 JUL 71 14:15
CASE 3	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MPE=6.50	15 JUL 71 14:57
CASE 4	LETS2 O/S STAGED COMBUSTION CYCLE	F=12.5K	MRE=5.50	15 JUL 71 14:58
CASE 5	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=6.50	15 JUL 71 15:01
CASE 6	LETS2 O/S STAGED COMBUSTION CYCLE	F=5.0K	MRE=5.50	15 JUL 71 15:35
CASE 7	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=6.50	16 JUL 71 08:55
CASE 8	LETS2 O/S STAGED COMBUSTION CYCLE	F=2.5K	MRE=5.50	16 JUL 71 08:55

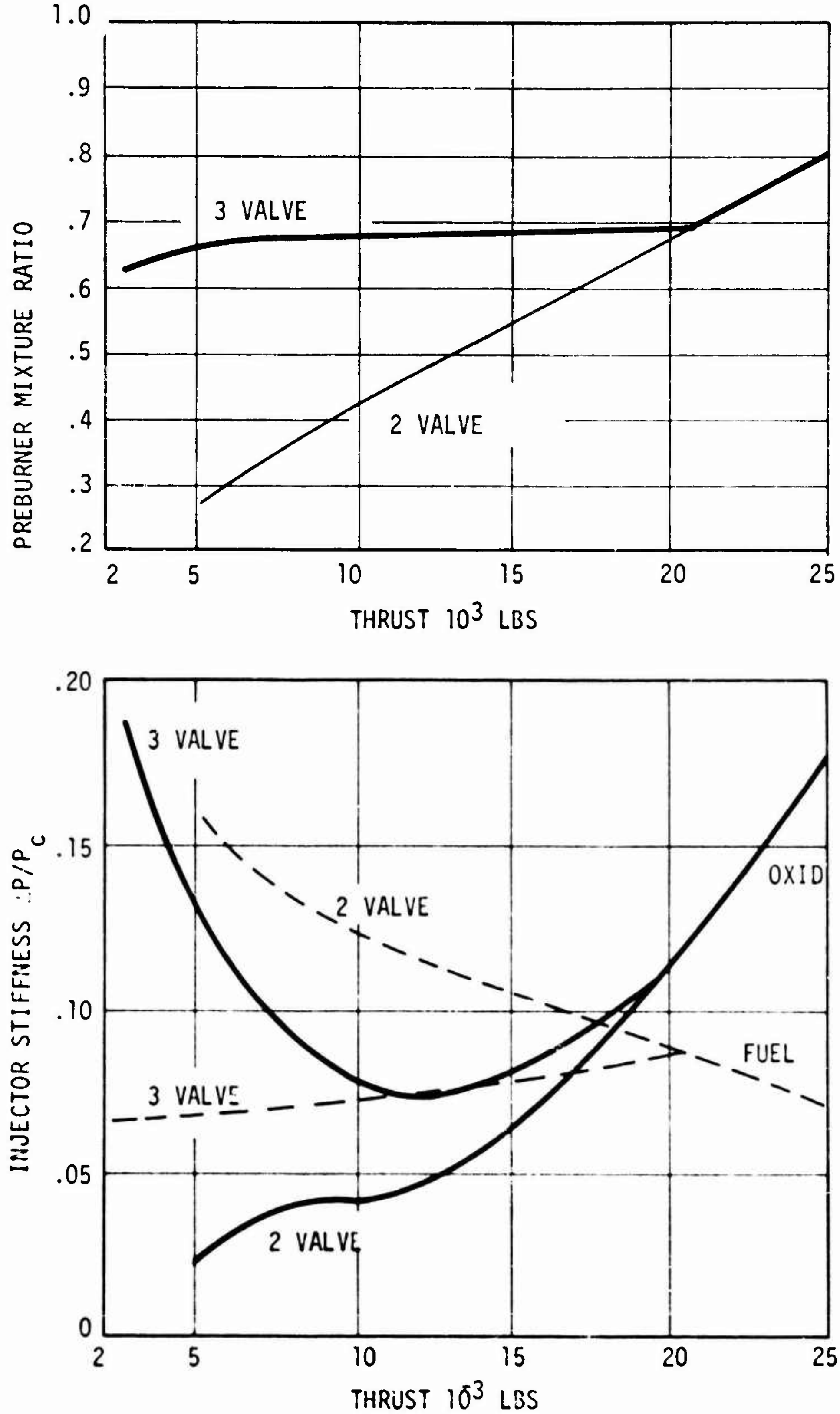


Figure 215. Preburner Mixture Ratio and Injector Stiffness 2- and 3-Valve Systems

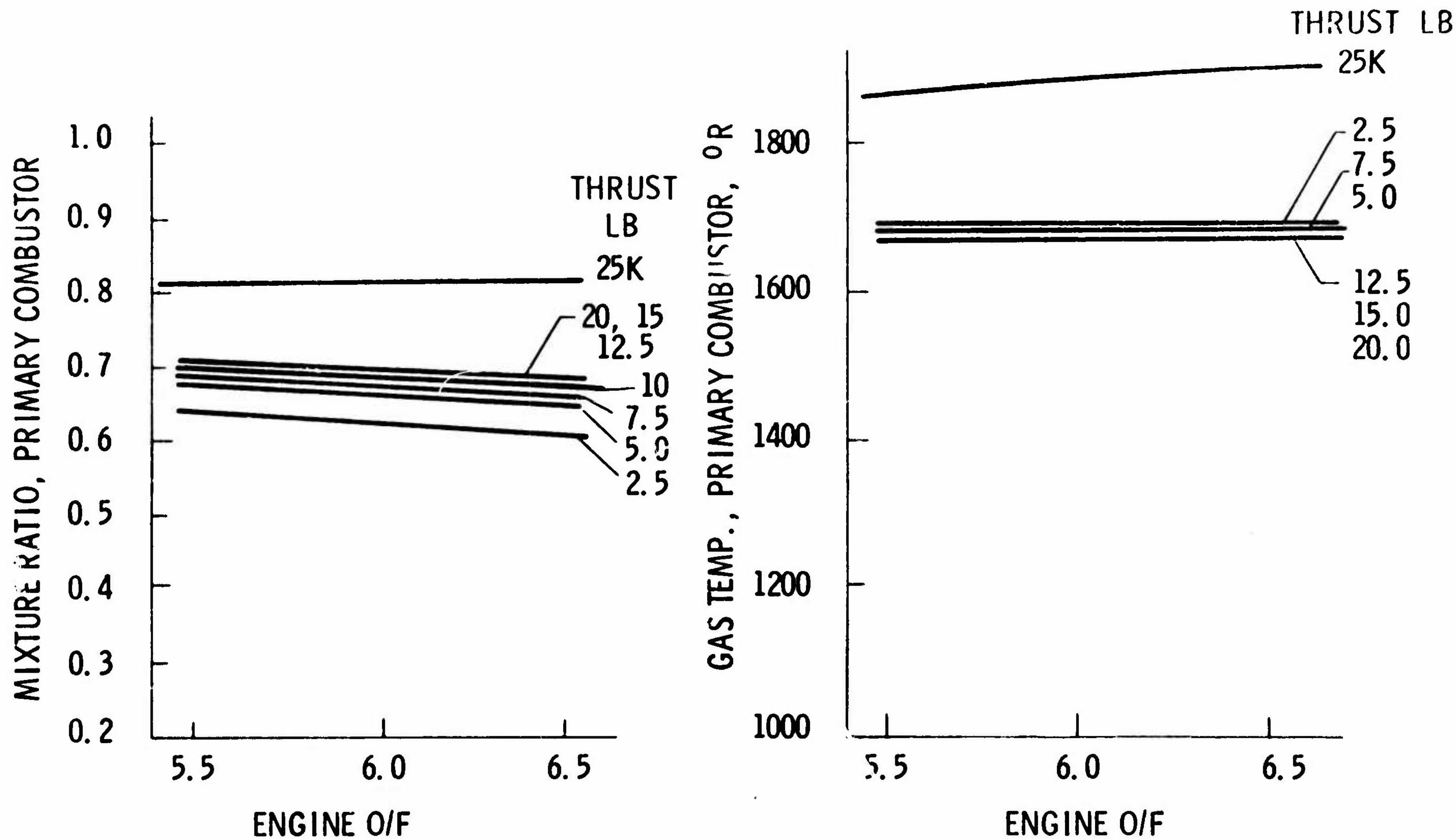


Figure 216. Primary Combustor MR and Gas Temperature of Off Design Operating Conditions (3 Valve)

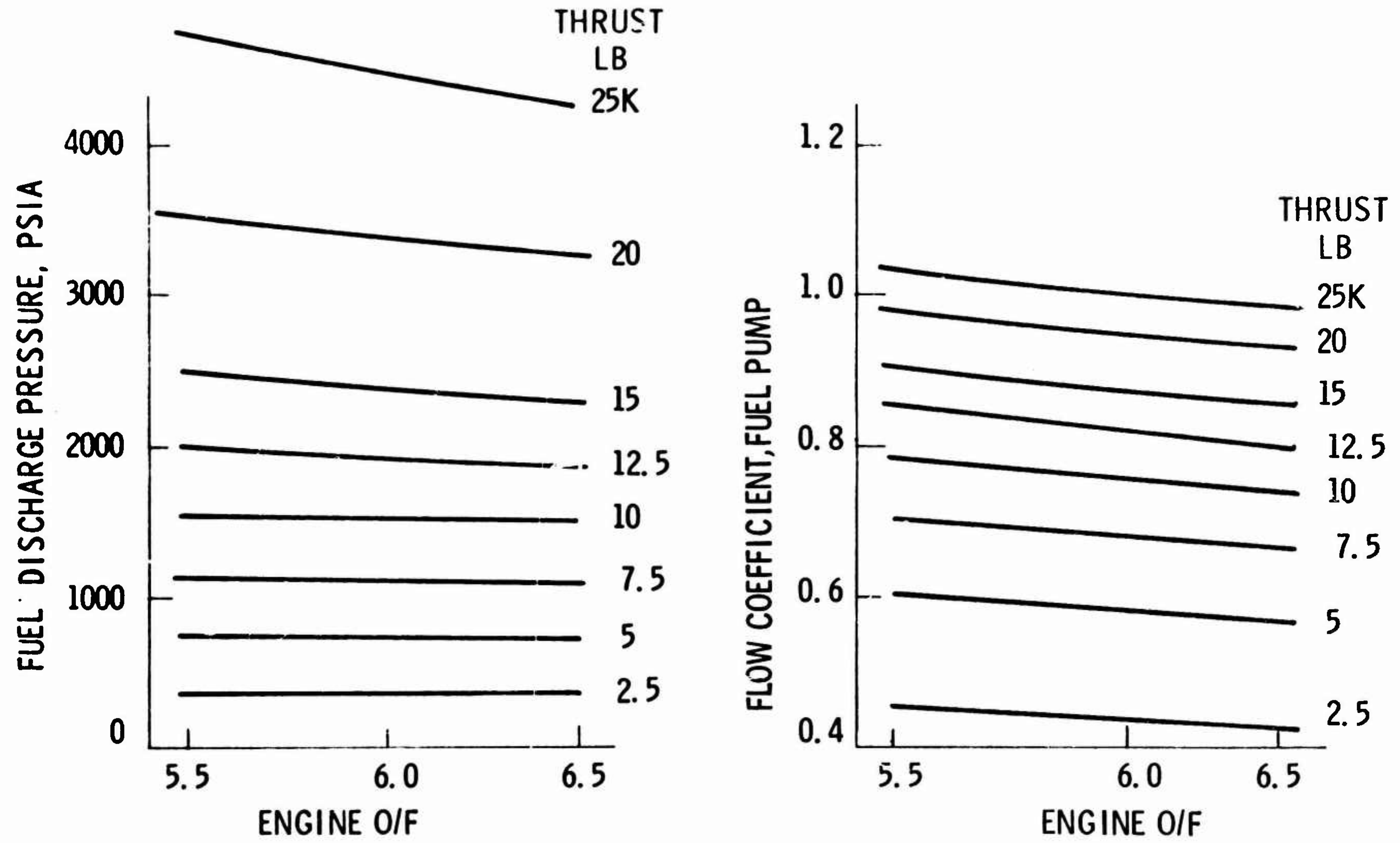


Figure 217. Fuel Pump Discharge Pressure and Flow Coefficient at Off Design Operating Conditions 10:1 Throttling (3 Valve)

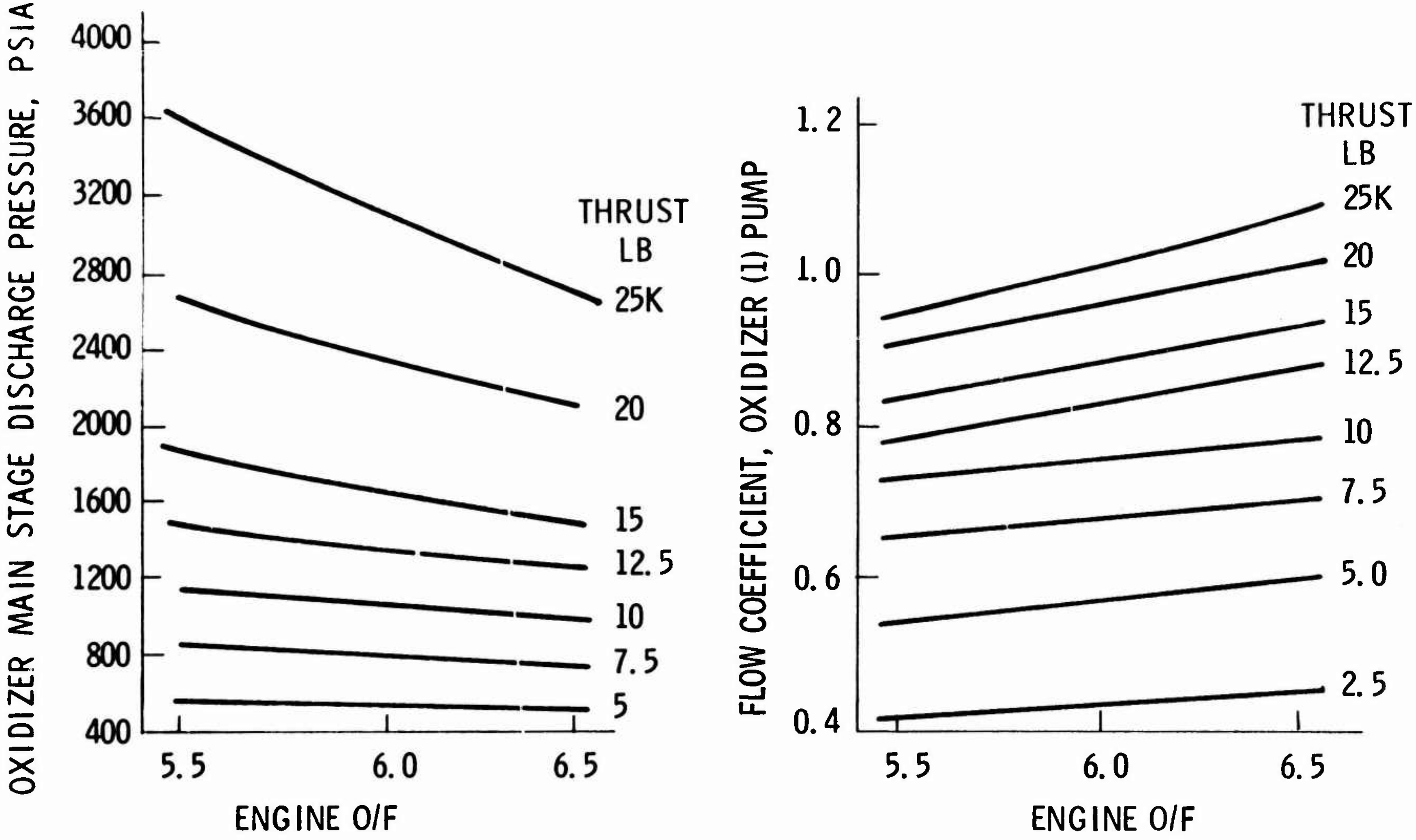


Figure 218. LO_2 Pump Discharge Pressure and Flow Coefficient at Off Design Operating Conditions 10:1 Throttling (3 Valve)

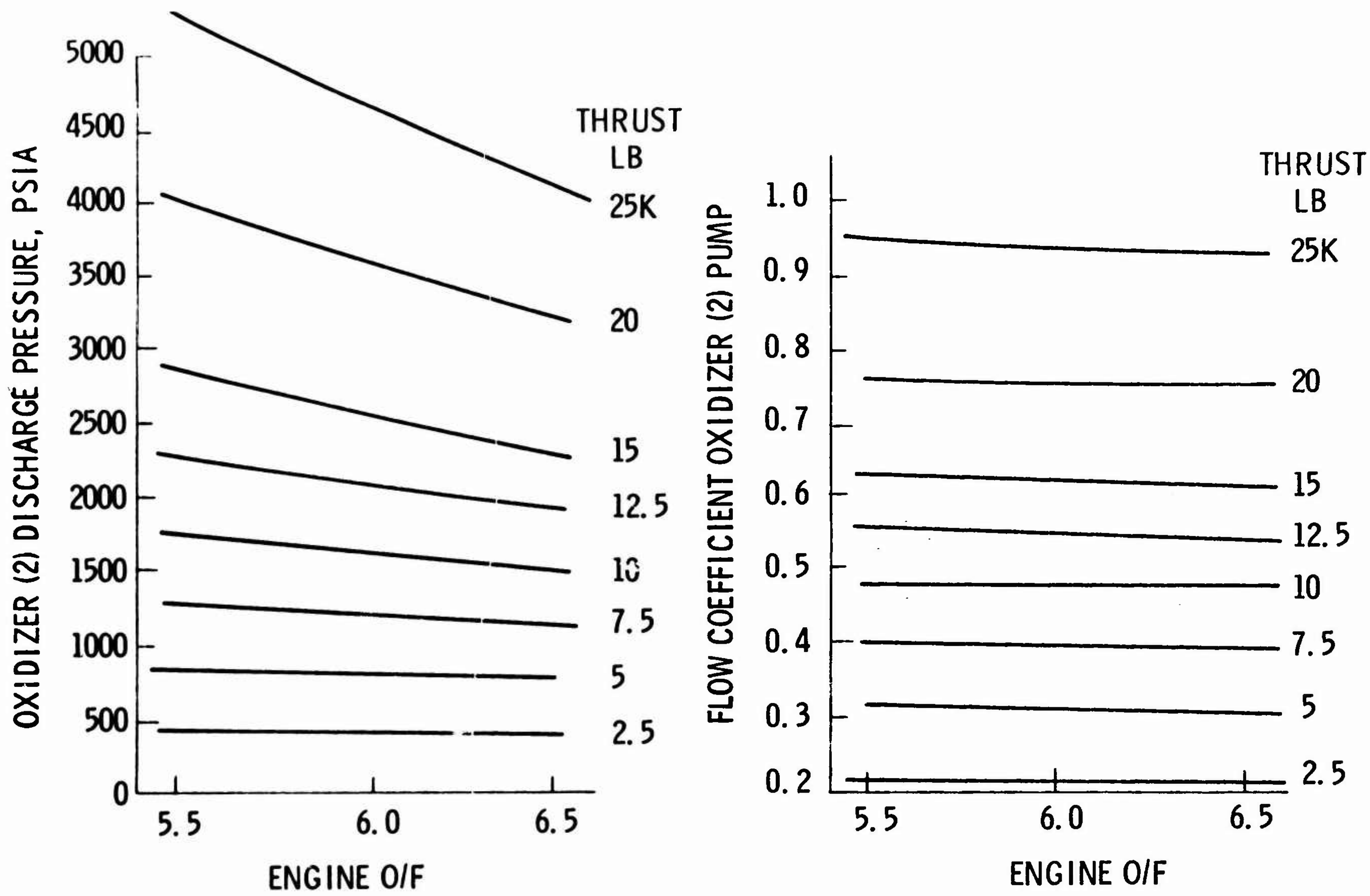


Figure 219. Pump Discharge Pressure at Off Design Operating Conditions 10:1 Throttling (3 Valve)

Main Oxidizer Pump
- Maximum flow coefficient 108% at 6.5, 25K
- Minimum discharge pressure 3600 psia at 5.5, 25K
- Maximum discharge pressure 3600 psia at 5.5, 25K

Half-Stage Oxidizer Pump
- Maximum flow coefficient 95% at 5.5, 25K
- Minimum flow coefficient 30% at 6.5, 5K
- Maximum discharge pressure 5302 psia at 5.5, 25K

Fuel and Oxidizer Gas Turbines -
- Maximum speeds, 83330/52577 rpm at 5.5, 25K
- Maximum gas temperature 1882°R at 6.5, 25K

Effect of Propellant Temperature

Figure 220 shows the effect on thrust and mixture ratio of variations in propellant enthalpy (constant tank pressure) for an engine operating at nominal (25K, MR = 6) with no feedback control for thrust or mixture ratio. Figure 221 shows the turbine inlet temperature with and without thrust and mixture ratio feedback control.

The purpose of this analysis is to demonstrate engine sensitivity to LO_2 and fuel inlet temperature conditions under full thrust conditions.

c. Engine Start and Shutdown Analysis

(1) Introduction

Upon completion of the steady state analysis, an engine start transient simulation was performed. The purpose of this analysis was: (1) identify possible problem areas during the start and determine solutions, (2) establish a preliminary valve sequence, (3) predict performance (thrust, impulse, and propellant consumption) for a nominal start.

The method used was to extend the LETS2 steady state model to provide for transient operation and then to "work through" an engine start. This is done by making an initial assumption as to valve sequencing and opening times then running 5 to 10 minutes on the computer. This will provide about 0.5 to 1.0 sec of engine transient. During this run, tape dumps are taken at intervals of 0.1 sec (engine time). The results are then examined to determine whether or not they are satisfactory and whether any problems exist. If the transient is not satisfactory, it is restarted from one of the tape dumps, changing valve timing from that point. This is a trial and error process and consumes considerable computer time, however, once it has been completed, it is then generally possible to run variations on the basic transient without the need for restarting the run.

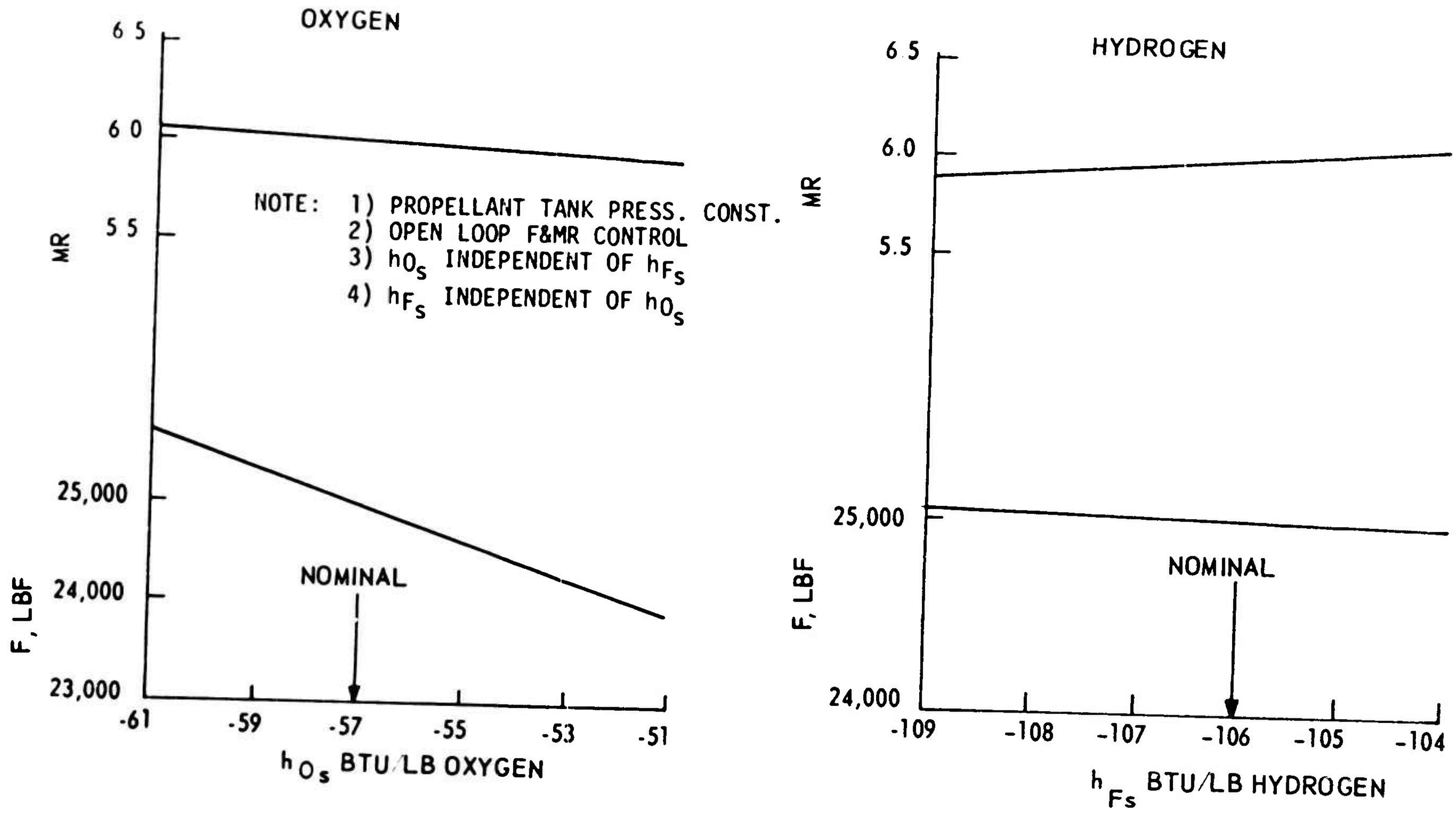


Figure 220. Effect of Propel'ant Inlet Enthalpy on Thrust and MR (MR = 6.0)

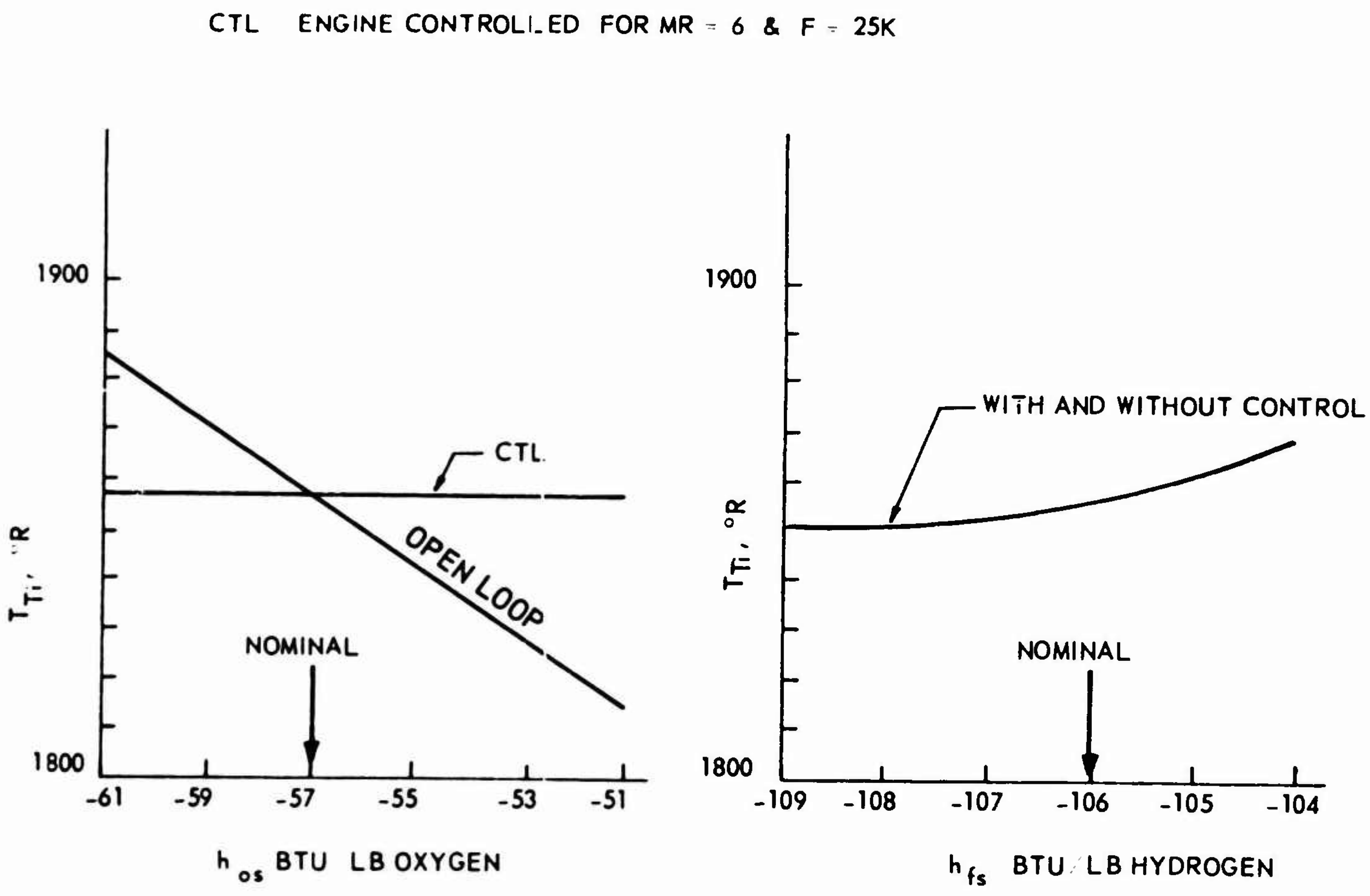


Figure 221. Turbine Inlet Temp Effect Due to Propellant Enthalpy

For this preliminary start sequence, fluid capacitance and inertia in the lines were neglected, permitting the use of a large (0.01 sec) computing interval for the LET2 solution. Inertias of the rotating machinery and the thermal capacitances of the cooling jacket and of the main and preburner oxidizer injectors are included in the model. The remainder of the components modeled for the LETS2 transient simulation is the same as that used for the LETS2 steady state model. The engine control system included a preburner fuel bypass valve as used on the 10:1 throttling version of the steady state model. Use of this valve during the start provides a method of separating control of preburner mixture ratio and turbine power. Thus, preburner mixture ratio can be held in the range from 0.6 to 0.9 giving reliable ignition and desirable warming of the oxidizer injectors, while holding back the rate of pump speed increase to minimize thrust chamber mixture ratio excursions and also possible cavitation problems associated with rapid pump speed buildup.

Figures 222 through 225 show the final predicted start transient. Figure 222 shows the valve timing and position used. Summarized start and shutdown performance data appear in Table LIX. Detailed discussion of various sections of the start transient appear below. The transient shown is not necessarily optimum, rather, it represents a nominal starting point for the development of an optimum transient.

The start transient may be divided into four sections which are described separately below. These are:

1. Chilldown and idle mode
2. Initial start - ignition and pump breakaway
3. Acceleration to 10% thrust level
4. Acceleration from 10% to full thrust

(2) Chilldown and Idle Mode

For the preliminary start transient, it was assumed that the start sequence was initiated with the pumps chilled down but with uncooled (560°R) metal in the cooling jacket and oxidizer injectors.

(3) Initial Start - Ignition and Pump Breakaway

The most significant factor during this part of the transient is the start of pump rotation. This is controlled by the static "breakaway" torques of the fuel and oxidizer turbopumps, and by the stall torque developed by the gas turbines. For the transient shown in Figure 219, the engine was started with the main oxidizer valve closed and with a preburner mixture ratio of approximately 0.8. Table LX shows a comparison of the turbine torques developed for this case and for three idle mode cases of interest. It can be seen that with design breakaway torques of 1 ft-lb for each turbopump assembly, satisfactory starts should be possible in the

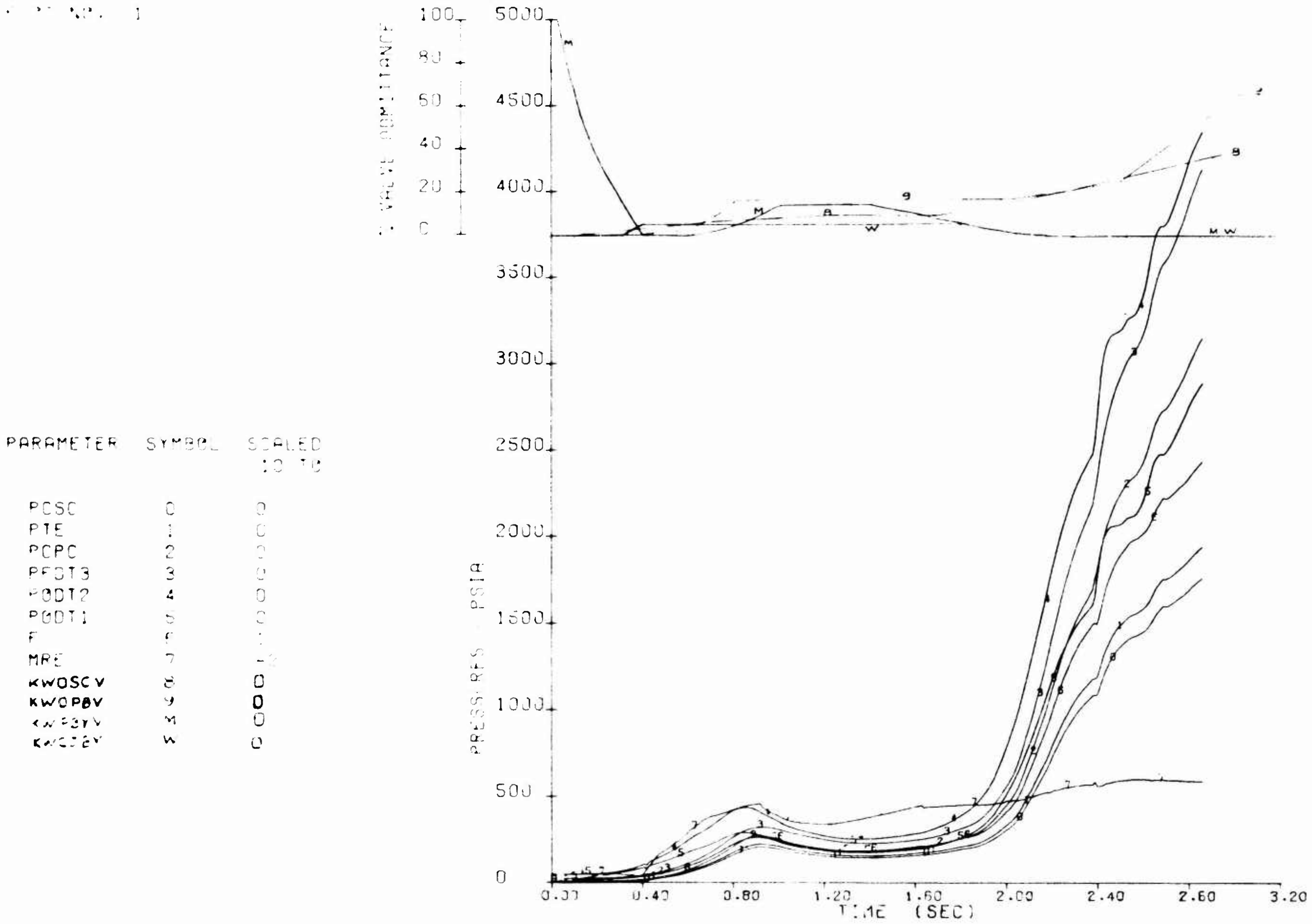


Figure 222. Start Transient Plot 1

PLOT NO. 3

PARAMETER	SYMBOL	SCALED 10 TO
F	0	1
PCSC	1	0
ITOT	2	1
WOTOT	3	-2
WFTOT	4	-2
MRE	5	-2
MRPC	6	-3
WOB	7	-2
WFB	8	-2
WTF	9	0
WTO	M	0

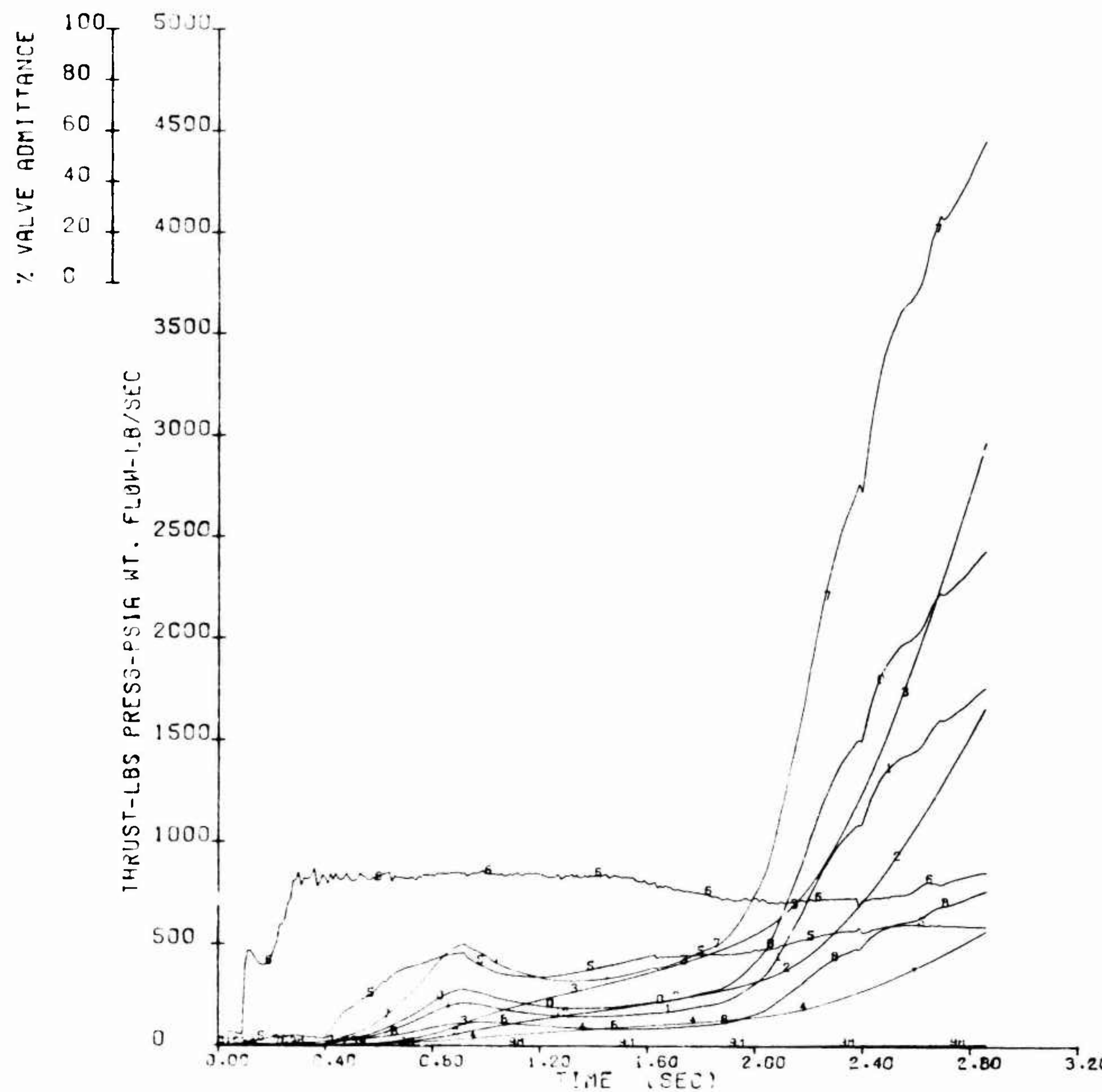


Figure 223. Start Transient Plot 3

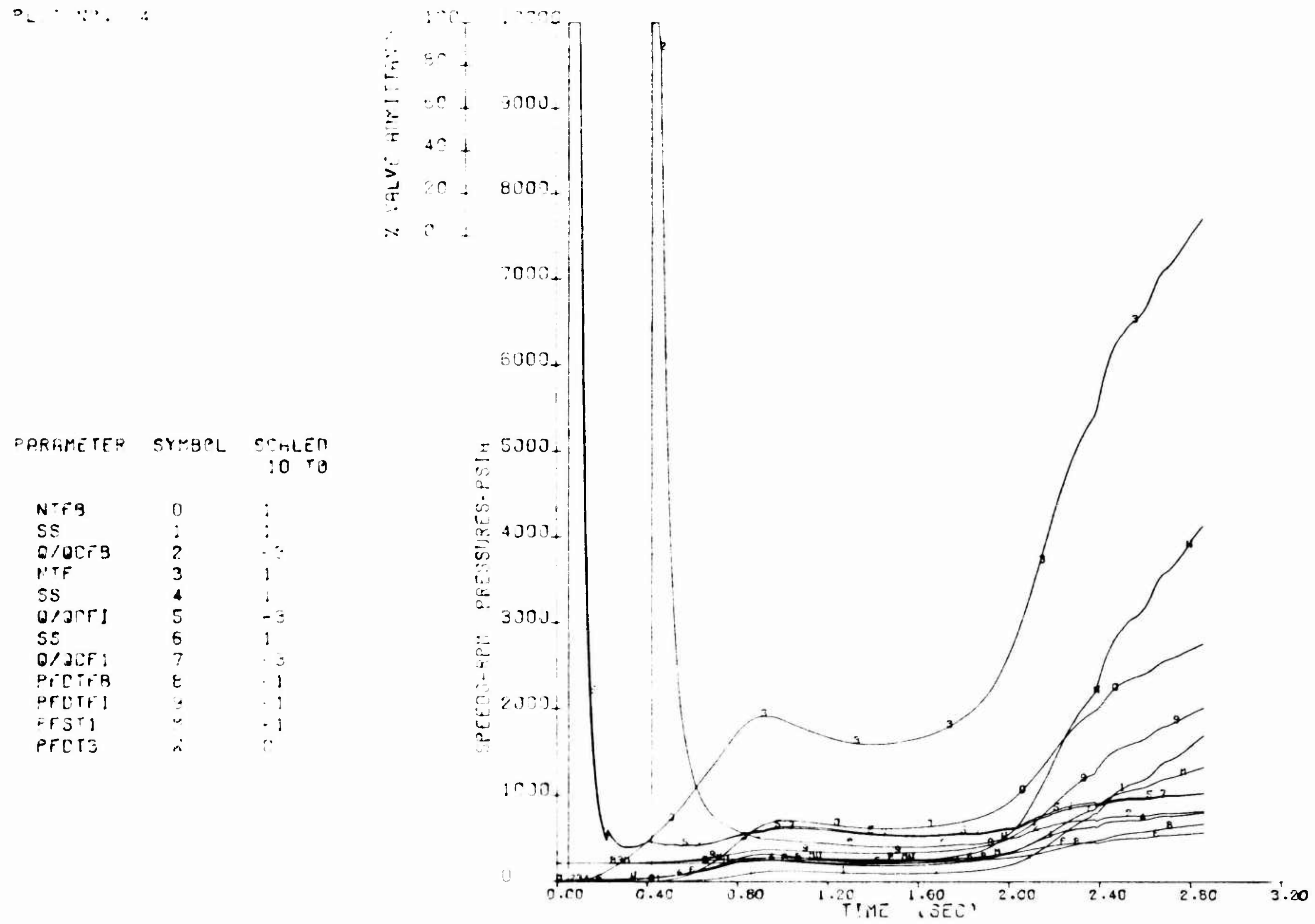


Figure 224. Start Transient Plot 4

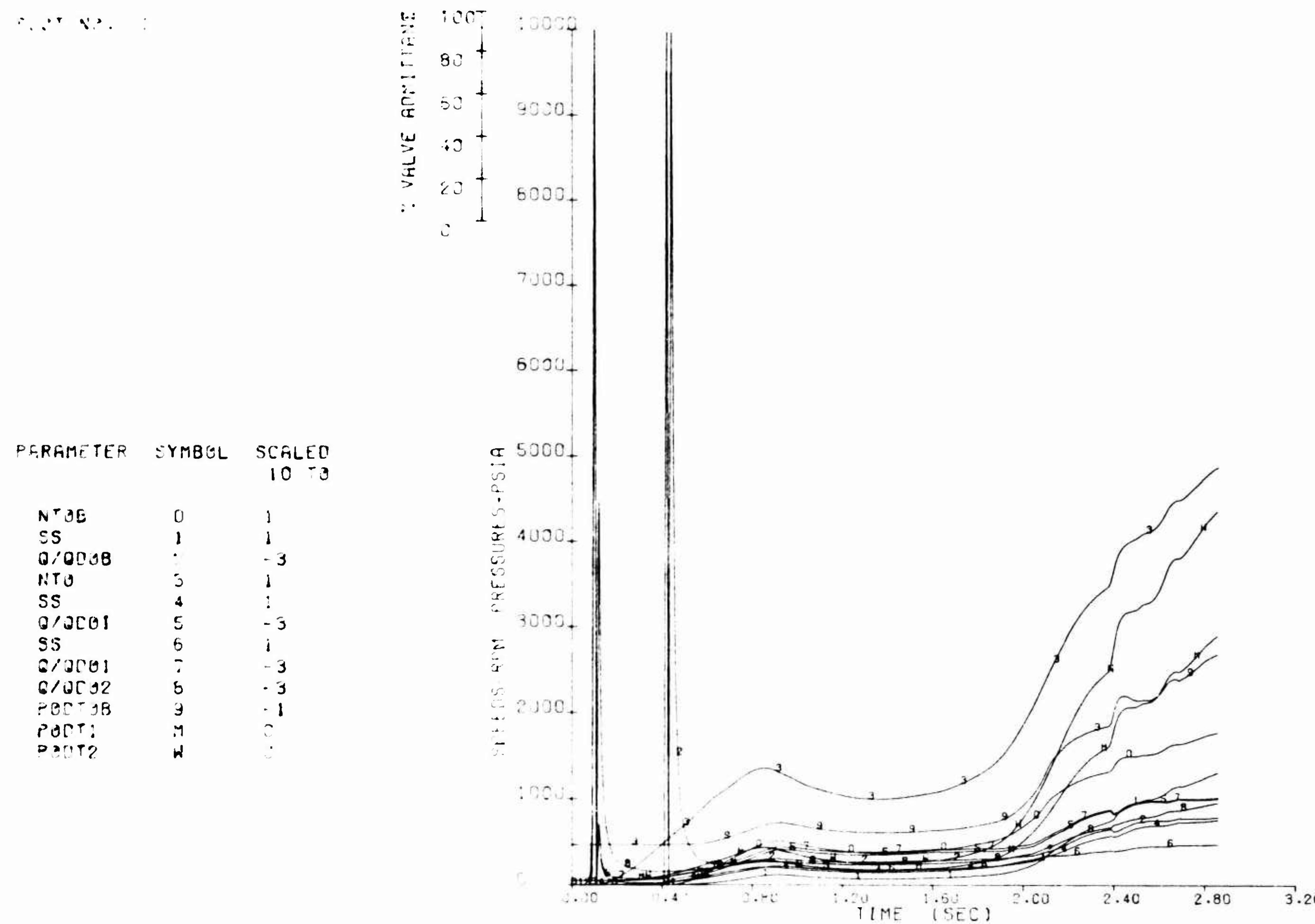


Figure 225. Start Transient Plot 5

TABLE LIX

TRANSIENT ANALYSIS SUMMARY

Engine Start Time	3.0 sec	Shutdown Time	1.5 sec
Start Impulse	18,780 lb/sec	Shutdown Impulse	9500 lb/sec
I_{sp} Avg.	440 sec	I_{sp} Avg.	400 sec
Propellant Consumed	46.82 lb	Propellant Consumed	23 lb
Thrust Rise Rate	37,000 lb/sec		

TABLE LX

TURBINE STALL TORQUE

CASE NO.	1	2	3	4
Engine Thrust, lb	30	106	96	96
Engine Mixture Ratio	0.81	4.11	2.24	3.11
Preburner Pressure, psia	17.4	17.4	11.3	10.8
Turbine Exhaust, psia	4.2	8.9	8.6	8.2
Preburner Mixture Ratio	0.80	0.78	0.06	0.07
Cooling Jacket Exit Temp., °R	283	283	272	508
Turbine Inlet Temp, °R	1636	1575	365	640
Fuel Turbine Torque, ft-lb	2.93	2.09	1.10	1.07
Oxid. Turbine Torque, ft-lb	1.68	1.20	0.45	0.44

NOTES:

All cases have preburner igniter on; fuel cooling jacket and preburner bypass valves closed.

CASE 1 - Staged combustion, no main oxidizer.
CASE 2 - Staged combustion, idle mode.
CASE 3 - Expander mode, 520°R cooling jacket.
CASE 4 - Expander mode, thermal equilibrium for pressure-fed idle.

pressure-fed idle mode as well as with no oxidizer flow, provided that the preburner is ignited. The two idle mode cases without preburner ignition show that lower breakaway torques would be required for repeatable and predictable starts in an expander mode.

(4) Acceleration to 10% Thrust Level

This portion of the transient introduces four major problem areas:

1. Pump stall
2. Preburner ignition
3. Pump cavitation
4. Oxidizer injector two-phase flow

All of these problems are related to the rate of thrust buildup required from the engine. The planned 3-sec transient seems a good choice in this respect. The plotted transient shows a dwell period from about 0.8 to 1.6 sec. This was introduced to provide more time for warming up the preburner and main oxidizer injectors. Careful examination of the results indicates that such a dwell period is probably not required and that the transient could be smoothed out in this region.

Pump stall will occur during this part of the transient because of the hydraulic characteristics of the engine. On the fuel pump, it will be controlled by opening the cooling jacket bypass (chilldown) valve sufficiently to increase the pump flow coefficient to a value greater than 0.5. This valve is sized for chilldown flow with gaseous hydrogen, in this application it will be flowing liquid hydrogen at about 5% of its wide open admittance. No special provision is made for control of oxidizer pump stall. It is minimized by keeping the buildup rate low (the oxidizer TPA tends to increase its speed faster than the fuel TPA).

Reliable preburner ignition required a mixture ratio greater than 0.6. Use of a preburner fuel bypass valve makes it possible to maintain preburner mixture ratio independently of turbine power and thus to provide satisfactory control of both buildup rate and mixture ratio.

Pump cavitation can occur due to dropping suction line pressures with rapid flow acceleration, or from too much lag in the boost pump speed buildup, or as a result of pump operation at extreme off-design flow coefficients. All of these are minimized by use of a relatively slow transient.

Two-phase flow may occur in the oxidizer injectors if the chamber pressure builds up faster than the oxidizer temperature in the injectors. It is not certain that this would be a problem, but the rapid changes in acoustic velocity and flow rate in a two-phase mixture have the potential of producing a high frequency oscillation. This was not investigated

at this time because the model used did not incorporate fluid capacitances and thus would not have modeled the high frequency (200-2000 Hz) range. Instead, the transient was controlled to keep both the main and preburner oxidizer injectors out of the two-phase region. This was accomplished by means of:

1. A relatively slow transient, providing time for injector warm up.
2. The hot preburner, resulting from the use of the preburner fuel bypass valve.
3. Keeping the engine mixture ratio on the low side throughout the transient thus reducing oxidizer flow rate.

(5) Acceleration from 10% to Full Thrust

No major problems were found in this region. As usual with fuel-rich staged combustion cycles, there will be a tendency to go oxidizer rich on acceleration. This is controlled by restricting the acceleration rate and by phasing the main oxidizer valve. One point to be noted is that there are several thermal capacitances included in the engine and if it is operated without feedback controls, these will probably produce a drift in the operating point over a period measured in seconds. This was not further investigated but should be if an open loop type of operation is selected.

(6) Engine Shutdown

No shutdown transient was run with the LETS-2 model. No problems are anticipated on shutdown. Closing the preburner oxidizer valve will cut turbine power and also produce a shift towards a lower engine mixture ratio. If required, the preburner fuel bypass valve can also be opened to give a more rapid shutdown.

d. Engine Stability Analysis

An analysis was performed to verify the dynamic stability of the engine cycle chosen for the Task I engine. Past experience has shown that instability may occur when throttleable engines are run at thrust levels less than 100-percent. Components are usually sized and designed to operate at the full-thrust level. Occasionally, this practice results in drastically reduced injector stiffnesses and/or significantly higher turbomachinery efficiencies when the engine is run at a lower thrust level. Also, care must be taken in designing the cycle such that the preburner never operates at a point where small perturbations in mixture ratio can cause large changes in enthalpy delivered to the turbines. The cooling jacket heat transfer design must be checked to insure that it does not sustain large pressure oscillations as the bulk transport properties change on each side of the tube wall during throttling. Each of the areas mentioned above can amplify or prolong the duration of disturbances in the system. Therefore, if the engine cycle appeared to be unstable, the system performance parameters

could be examined to discover which component or combination of components was causing the instability. Component design changes could then be recommended to rectify the situation.

In order to determine the dynamics of the engine cycle, a mathematical model of the engine was created. The propellant circuits were described using turbulent compressible flow equations. Liquid inertances and capacitances were included in the propellant feed lines. Pump inlet capacitances were also considered. Rotating machinery elements were described by the standard second order differential relationship

$$\Sigma T = I \frac{d^2\theta}{dt^2}$$

Where ΣT is the sum of the torques acting on the pump shaft, I is the total polar moment of all rotating components (turbine, pump impeller, shaft, bearings, seals, etc.), and θ is the angular displacement of the shaft. In the regenerative sections of both primary and secondary combustors, bulk fluid temperature was used to calculate the propellant heat transfer properties. The relationships described above formed a non-linear set of equations which modeled the operating performance of the various engine components. That set of equations was then linearized such that small perturbation theory could be applied. The linear analysis permitted the use of existing computer algorithms. The linear analysis was less costly and shorter duration than non-linear methods.

The linearized math model of the engine was constructed and is graphically represented in Figure 226. The bond graph was used to establish a sign convention for the set of equations. The arrow heads on the bonds indicate the direction of power flow through the cycle. A half-head symbolizes a passive bond where information is transmitted in both directions across the bond. A full-head indicates that the bond is active, and signals can only be transmitted in the direction of the arrow. The perpendicular bar at the extremity of a bond is termed a "causal stroke". The "causal stroke" is placed on the end of the bond from whence the flow signal is coming. The "causal strokes" were assigned to the bond graph such that integral causality would result. The dynamic elements in the model are inductors and capacitors. By assigning integral causality, the inductors generate flows and the capacitors generate pressures such that a set of integral equations will be evolved.

In order to verify engine cycle stability, the engine component equations were linearized at mixture ratio of six and three thrust levels: 100%, 50%, and 20%. The $\underline{A}$-matrix coefficients for the three conditions are shown in Table LXI. The eigenvalues for the three thrust levels are shown in Table LXII. In each case, the computer calculated eigenvalues, all of which had negative real parts. Therefore, at each thrust level the roots of the characteristic equation are negative, indicating that disturbances will attenuate with time. Had any eigenvalues been calculated

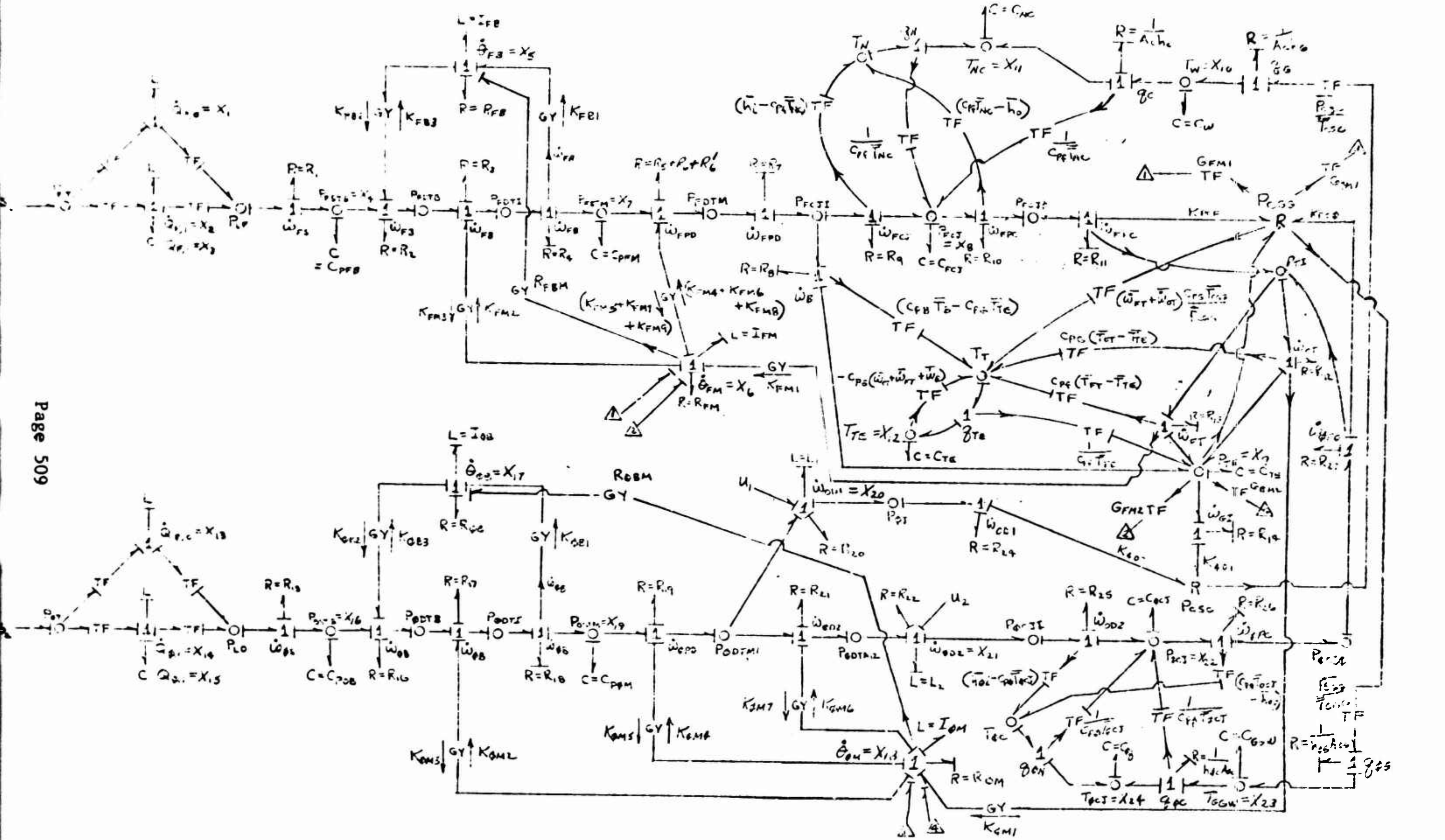


Figure 226. Bond Graph of Linearized Engine Model

TABLE LXI

MATRIX FOR THE ENGINE SYSTEM

Element	F = 25K	F = 12.5K	F = 5K
A(1, 1) =	-.31669+02	-.31669+02	-.31669+02
A(1, 2) =	.31669+02	.31669+02	.31669+02
A(1, 4) =	-.28813+03	-.28813+03	-.28813+03
A(2, 1) =	.31669+02	.31669+02	.31669+02
A(2, 2) =	-.31669+02	-.31669+02	-.31669+02
A(2, 3) =	-.29106+05	-.29106+05	-.29106+05
A(2, 4) =	.28813+03	.28813+03	.28813+03
A(3, 2) =	.10000+01	.10000+01	.10000+01
A(4, 1) =	.16101+00	.21955+01	.24151+02
A(4, 2) =	-.16101+00	-.21955+01	-.24151+02
A(4, 4) =	-.27067+00	-.39942+01	-.59880+02
A(4, 5) =	-.99134-02	-.11924+00	-.11377+01
A(4, 6) =	-.13534-01	-.16776+00	-.14970+01
A(4, 7) =	.27067+00	.39942+01	.59880+02
A(5, 4) =	.35695+02	.41447+02	.11677+03
A(5, 5) =	-.53561+02	-.53565+02	-.12690+02
A(5, 6) =	.17615+02	.17574+02	.56358+01
A(5, 7) =	-.35695+02	-.41447+02	-.11677+03
A(6, 4) =	-.28711+01	-.34687+01	-.47274+01
A(6, 5) =	-.68948-01	-.99205-01	-.89820-01
A(6, 6) =	-.10778+03	-.73987+02	-.55683+02
A(6, 7) =	-.28860+02	-.35536+02	-.42257+02
A(6, 8) =	.15205+03	.59021+02	.11586+03
A(6, 9) =	-.33802+03	-.62845+03	-.43879+03
A(6,22) =	.12929+03	.10020+02	.29350+03
A(7, 4) =	.46631+02	.43937+02	.59880+02
A(7, 5) =	.13520+01	.12566+01	.11377+01
A(7, 6) =	-.16849+01	-.80420+00	-.34831+00
A(7, 7) =	-.43376+02	-.47025+02	-.63187+02
A(7, 8) =	.27627+01	.30811+01	.32965+01
A(7, 9) =	.61010-02	.69040-02	.66515-02
A(8, 6) =	.45753+01	.12553+01	-.70847+00
A(8, 7) =	.40702+01	.14631+01	-.12697+01
A(8, 8) =	-.39222+02	-.507[illegible]	-.53093+02
A(8, 9) =	-.25944+02	-.21560+02	-.21171+02
A(8,10) =	.86139+02	.75394+02	.83670+02
A(8,11) =	-.86139+02	-.75894+02	-.83870+02
A(8,22) =	.91643-01	.27762+02	.33200+03
A(9, 6) =	.27356-01	.19534-01	.11277-01
A(9, 7) =	.24517-01	.22737-01	.20210-01
A(9, 8) =	.10704+03	.85320+02	.57523+02
A(9, 9) =	-.81413+03	-.62335+03	-.54910+03
A(9,12) =	-.37141+03	-.16700+03	-.67440+02
A(9,20) =	.15133+05	.65011+04	.74351+04
A(9,22) =	.23391+03	.26065+03	.24946+03
A(10, 9) =	.14129+01	.29747+01	.75842+01
A(10,10) =	-.33715+01	-.33715+01	-.33715+01
A(10,11) =	.28165+01	.28155+01	.28165+01
A(10,20) =	.15235+02	.24429+02	.53207+02
A(11, 6) =	-.21974+01	-.34874+01	-.84220+01
A(11, 7) =	-.19533+01	-.40645+01	-.15093+02
A(11, 8) =	.13529+01	.21027+01	.79159+01
A(11, 9) =	.41837+00	.88914+00	.27341+01
A(11,10) =	.86942+01	.11620+02	.35531+02
A(11,11) =	-.86942+01	-.11620+02	-.35531+02
A(11,22) =	.15150+00	.11128+01	.44133+01
A(12, 6) =	-.26331+00	[illegible]	

(Cont.)

F = 25K		F = 12.5K		F = 5K	
A(12, 7) =	-.25806+00	A(12, 7) =	-.24450	A(12, 7) =	-.58027+00
A(12, 8) =	.13740+03	A(12, 8) =	.29503	A(12, 8) =	.37121+02
A(12, 9) =	.82104+02	A(12, 9) =	.67380+02	A(12, 9) =	.99755+02
A(12,12) =	-.33145+03	A(12,12) =	-.16749+03	A(12,12) =	-.14211+03
A(12,22) =	.28717+02	A(12,22) =	.66541+02	A(12,22) =	.15470+03
A(13,13) =	-.15673+01	A(13,13) =	-.15673+01	A(13,13) =	-.15673+01
A(13,14) =	.15673+01	A(13,14) =	.15673+01	A(13,14) =	.15673+01
A(13,16) =	-.54401+02	A(13,16) =	-.54401+02	A(13,16) =	-.54401+02
A(14,13) =	.15673+01	A(14,13) =	.15673+01	A(14,13) =	.15673+01
A(14,14) =	-.15673+01	A(14,14) =	-.15673+01	A(14,14) =	-.15673+01
A(14,15) =	-.23258+06	A(14,15) =	-.23258+06	A(14,15) =	-.23253+06
A(14,16) =	.54401+02	A(14,16) =	.54401+02	A(14,16) =	.54401+02
A(15,14) =	.10000+01	A(15,14) =	.10000+01	A(15,14) =	.10000+01
A(16,13) =	.24916+02	A(16,13) =	.32582+02	A(16,13) =	.42357+02
A(16,14) =	-.24916+02	A(16,14) =	-.32582+02	A(16,14) =	-.42357+02
A(16,16) =	-.35651+01	A(16,16) =	-.47483+01	A(16,16) =	-.80645+01
A(16,17) =	-.65954+00	A(16,17) =	-.97361+00	A(16,17) =	-.11532+01
A(16,18) =	-.13084+01	A(16,18) =	-.15432+01	A(16,18) =	-.14194+01
A(16,19) =	.35651+01	A(16,19) =	.47483+01	A(16,19) =	.80645+01
A(17,16) =	.65455+02	A(17,16) =	.63704+02	A(17,16) =	.66371+02
A(17,17) =	-.26239+03	A(17,17) =	-.14394+03	A(17,17) =	-.66109+02
A(17,18) =	.12652+03	A(17,18) =	.79304+02	A(17,18) =	.26581+02
A(17,19) =	-.65455+02	A(17,19) =	-.63704+02	A(17,19) =	-.66371+02
A(18, 8) =	.33711+02	A(18, 8) =	.23553+02	A(18, 8) =	.20698+02
A(18, 9) =	-.95511+02	A(18, 9) =	-.12299+03	A(18, 9) =	-.13257+03
A(18,16) =	-.19284+02	A(18,16) =	-.74822+01	A(18,16) =	-.58956+01
A(18,17) =	-.55675+01	A(18,17) =	-.15359+01	A(18,17) =	-.84308+00
A(18,18) =	-.49123+02	A(18,18) =	-.19326+02	A(18,18) =	-.14674+02
A(18,19) =	.19284+02	A(18,19) =	.74822+01	A(18,19) =	.58956+01
A(18,20) =	-.24167+03	A(18,20) =	-.34091+03	A(18,20) =	-.45455+03
A(18,21) =	-.4[illegible]+03	A(18,21) =	-.10417+04	A(18,21) =	-.10727+04
A(18,22) =	.37028+02	A(18,22) =	.70650+02	A(18,22) =	.87629+02
A(19,16) =	.55598+01	A(19,16) =	.[illegible]1748+01	A(19,16) =	.57204+01
A(19,17) =	.10195+01	A(19,17) =	.12654+01	A(19,17) =	.96102+00
A(19,18) =	.20220+01	A(19,18) =	.20062+01	A(19,18) =	.11828+01
A(19,19) =	-.55098+01	A(19,19) =	-.61728+01	A(19,19) =	-.67204+01
A(19,20) =	-.93909+02	A(19,20) =	-.10000+03	A(19,20) =	-.83333+02
A(19,21) =	-.93909+02	A(19,21) =	-.10000+03	A(19,21) =	-.83333+02
A(20, 9) =	-.19485+02	A(20, 9) =	-.19438+02	A(20, 9) =	-.19962+03
A(20,18) =	.31915+02	A(20,18) =	.36385+02	A(20,18) =	.13298+02
A(20,19) =	.26596+02	A(20,19) =	.26598+02	A(20,19) =	.26595+02
A(20,20) =	-.26164+04	A(20,20) =	-.25160+04	A(20,20) =	-.18687+04
A(20,21) =	-.7[illegible]+03	A(20,21) =	-.11170+04	A(20,21) =	-.53191+03
A(21,18) =	.42[illegible]+01	A(21,18) =	.43434+01	A(21,18) =	.21278+01
A(21,19) =	.25253+01	A(21,19) =	.25253+01	A(21,19) =	.25253+01
A(21,20) =	-.70707+02	A(21,20) =	-.10606+03	A(21,20) =	-.59505+02
A(21,21) =	-.62323+03	A(21,21) =	-.26212+04	A(21,21) =	-.49785+04
A(21,22) =	-.25253+01	A(21,22) =	-.25253+01	A(21,22) =	-.25253+01
A(22, 8) =	.52771+00	A(22, 8) =	.11590+00	A(22, 8) =	.61313-01
A(22, 9) =	.39795+00	A(22, 9) =	.26047+00	A(22, 9) =	.21296+00
A(22,21) =	-.11[illegible]+03	A(22,21) =	-.64651+02	A(22,21) =	-.30755+02
A(22,22) =	-.92856+00	A(22,22) =	-.37858+00	A(22,22) =	-.29378+00
A(22,23) =	.15905+01	A(22,23) =	.28060+00	A(22,23) =	.51310-01
A(22,24) =	-.15905+01	A(22,24) =	-.28060+00	A(22,24) =	-.51310-01

TABLE LXI (cont.)

	F = 25K	F = 12.5K	F = 5K
A(23, 8) =	.19418+01	.46817+00	.33643+00
A(23, 9) =	.14639+01	.10521+01	.87345+00
A(23,22) =	.51663+00	.13544+01	.13778+01
A(23,23) =	-.10411+02	-.45343+01	-.19602+01
A(23,24) =	.45000+01	.16206+01	.70551+00
A(24, 8) =	-.73343-01	-.61328-01	-.55002-01
A(24, 9) =	-.55293-01	[illegible]371+00	-.13782+[illegible]
A(24,21) =	-.49100+02	-.58199+02	-.47298+0[illegible]
A(24,22) =	.12864+00	.19672+00	.19915+00
A(24,23) =	.15171+00	.25854-01	.50864-01
A(24,24) =	-.15171+00	-.25854-01	-.50864-01

TABLE LXII

EIGEN VALUES FOR THE OOS ENGINE MODEL
LINEARIZED AT MR=6 AND THREE THRUST LEVELS

Engine Thrust = 25K, MR = 6

EIGEN VALUES

1	-.1153+000	-.3648+000	2	-.1153+000	.3648+000	3	-.3354+000	.0000
4	-.8032+000	-.4837+003	5	-.8032+000	.4837+003	6	-.1543+001	.0000
7	-.2391+001	-.3035+002	8	-.2391+001	.3035+002	9	-.6294+001	-.1035+001
10	-.6294+001	.1035+001	11	-.1029+002	.0000	12	-.1532+002	-.1671+003
13	-.1532+002	.1671+003	14	-.3112+002	.0000	15	-.4280+002	.0000
16	-.4838+002	-.4717+001	17	-.4838+002	.4717+001	18	-.6542+002	.0000
19	-.1164+003	.0000	20	-.2874+003	.0000	21	-.3977+003	.0000
22	-.8378+003	-.7952+002	23	-.8378+003	.7952+002	24	-.2490+004	.0000

F = 12.5K, MR = 6

EIGEN VALUES

1	-.1866+000	-.1492+000	2	-.1866+000	.1492+000	3	-.4296+000	.0000
4	-.6138+000	-.4641+003	5	-.6138+000	.4641+003	6	-.3201+001	-.4143+002
7	-.3201+001	.4143+002	8	-.4625+001	.0000	9	-.8645+001	.0000
10	-.1604+002	-.1687+003	11	-.1604+002	.1687+003	12	-.1679+002	-.1846+002
13	-.1679+002	.1846+002	14	-.2477+002	-.9301+001	15	-.2477+002	.9301+001
16	-.5100+002	-.8957+001	17	-.5100+002	.8957+001	18	-.6615+002	-.5653+001
19	-.6615+002	.5653+001	20	-.1417+003	.0000	21	-.2168+003	.0000
22	-.6044+003	.0000	23	-.2158+004	.0000	24	-.2885+004	.0000

F = 5K, MR = 6

EIGEN VALUES

1	-.3653-001	-.4426-001	2	-.3653-001	.4426-001	3	-.5087+000	.0000
4	-.6410+000	-.4847+003	5	-.6410+000	.4847+003	6	-.2031+001	.0000
7	-.5166+001	-.4712+002	8	-.5166+001	.4712+002	9	-.1014+002	.0000
10	-.1690+002	.0000	11	-.2003+002	-.5888+001	12	-.2003+002	.5888+001
13	-.2511+002	-.6413+002	14	-.2511+002	.6413+002	15	-.2858+002	-.1851+003
16	-.2858+002	.1651+003	17	-.6341+002	.0000	18	-.6758+002	-.4003+002
19	-.6758+002	.4003+002	20	-.8025+002	.0000	21	-.1611+003	.0000
22	-.6397+003	.0000	23	-.1729+004	.0000	24	-.4986+004	.0000

positive real parts, the engine cycle would reinforce spurrious disturbances and would be considered unstable.

The eigenvalues may be interpreted to obtain additional dynamic information about the cycle besides the stability verification. The highly underdamped oxidizer feed line sustains oscillations for several seconds. These flow oscillations strongly affect the speeds of the oxidizer pumps which feed the engine. Therefore, during throttling the engine thrust and mixture ratio would vary at the new thrust level until the system oscillations die out. This undesirable feature can be corrected by installing a capacitor at the ox boost pump inlet or by actively controlling the oxidizer throttle valves with a boost pump suction pressure signal.

At full thrust, the fuel feed line and oxidizer feed line resonate at 26.6 Hz and 77 Hz, respectively. Vehicle structural dynamic resonances should not coalesce with these frequencies in order to avoid the onset of pogo oscillations during flight. The line resonances can be modified to other frequencies by active or passive control devices if line-vehicle resonance matching is recognized during the interface design study.

The speed of response of the system is approximated by the inverse of the magnitude of a real root. The slowest poles of the engine system respond in 5 sec at 25K, 5.3 sec at 12.5K and 20 sec at 5K. The engine parameters which are affected the most by the slow response are the cooling jacket wall temperature and the bulk fluid temperature of the hydrogen in the main combustor regneerative nozzle. These parameters have a minor influence on engine thrust and mixture ratio, so the slow response of those parameters would not impact mission performance.

At the 20% thrust level, the lowest frequency pair of complex roots approaches the origin, resulting in extremely slow response of the system. Examination of the system eigenvectors indicates that the slow response attributed to the cooling jacket dynamics of the main chamber is caused by the heat capacitance of the tube bundle wall and the reduced flow of hydrogen through the tubes.

The linearized dynamic model of the OO% Engine cycle can be used in subsequent phases of this project to design the engine steady state controller for thrust and mixture ratio control. In addition, the model can provide dynamic information with respect to potential POGO problems associated with engine-vehicle interactions. Engine design changes may be dynamically evaluated using the model, and the other design points may be tested, such as other thrust levels and mixture ratios, as the detailed engine design evolves. The most significant finding of this study was the fact that the engine cycle is stable, so the engine could be controlled open loop. Since the oxidizer feed line oscillations persist for several seconds, a closed loop control scheme or a passive device should be incorporated to attenuate those oscillations.

4. Interface Requirements

The interface requirements section summarizes all interface data for the 25K OOS engine design for the purpose to have a ready reference section of the engine capability to be used for vehicle engine integration.

a. OOS Engine Interface Data

The OOS, a LO_2/LH_2 staged-combustion cycle engine, is shown in plan and side view on Figure 227. The hot gas manifold is the primary engine structural component. The high pressure turbopump assemblies (HPTPA's), including integrated in-line low pressure turbopump assemblies (LPTPA's), are side mounted laterally to an integrated turbine housing, as is the single preburner. The gimbal block assembly is affixed to the upper side of the conical centerbody and the injector, thrust chamber/nozzle assembly, H_2 regeneratively cooled to an exit area ratio, ε_o = 290:1, to the lower side. Additionally, the main combustion chamber igniter is mounted to the conical centerbody of the manifold.

Engine gimbaling is accommodated by articulating the propellant lines between the engine mounted boost pump inlets and the vehicle utilizing line routing options available to the vehicle contractor, as presented herein. Gimbal actuator lower attach clevises are located on the thrust chamber and cylindrical envelopes are provided for actuator clearance definition and engine packaging optimization. Hydrogen lines and turbopump upstream of the fuel discharge valve are insulated. The discharge line downstream of the discharge valve to the chamber coolant jacket inlet is also insulated to minimize heat input into the low flow hydrogen during engine idle mode operation.

Engine instrumentation, electrical harnesses and small propellant recirculating lines and helium purge lines are not shown in order to provide clarity of major component design integration into the engine assembly. Packaging of an engine mounted controller is limited to engine mounting location and estimated envelope definition because final controller design/size is dependent on establishment of final instrumentation and integrated circuitry with the vehicle.

Engine characteristic data are presented in Figure 228. Interface dimensions are shown on the engine assembly in Figure 227. The interim propellant line/vehicle interface shown provides flexibility to the vehicle contractor in selecting from suction line routing options presented in Figure 229 which best suit the constraints of the vehicle/engine combinations. Options presented include pressure volume compensating bellows (PVC's) and gimbaled suction line routings and interfaces. Engine gimbal envelope data are shown in Figure 230. Engine/vehicle envelope compatibility is demonstrated for a gimbal angle of $\pm$ 5° pitch and yaw. Engine weights are presented in Table LXIII. Engine weight charge as a function of NPSH is presented in Figure 231.

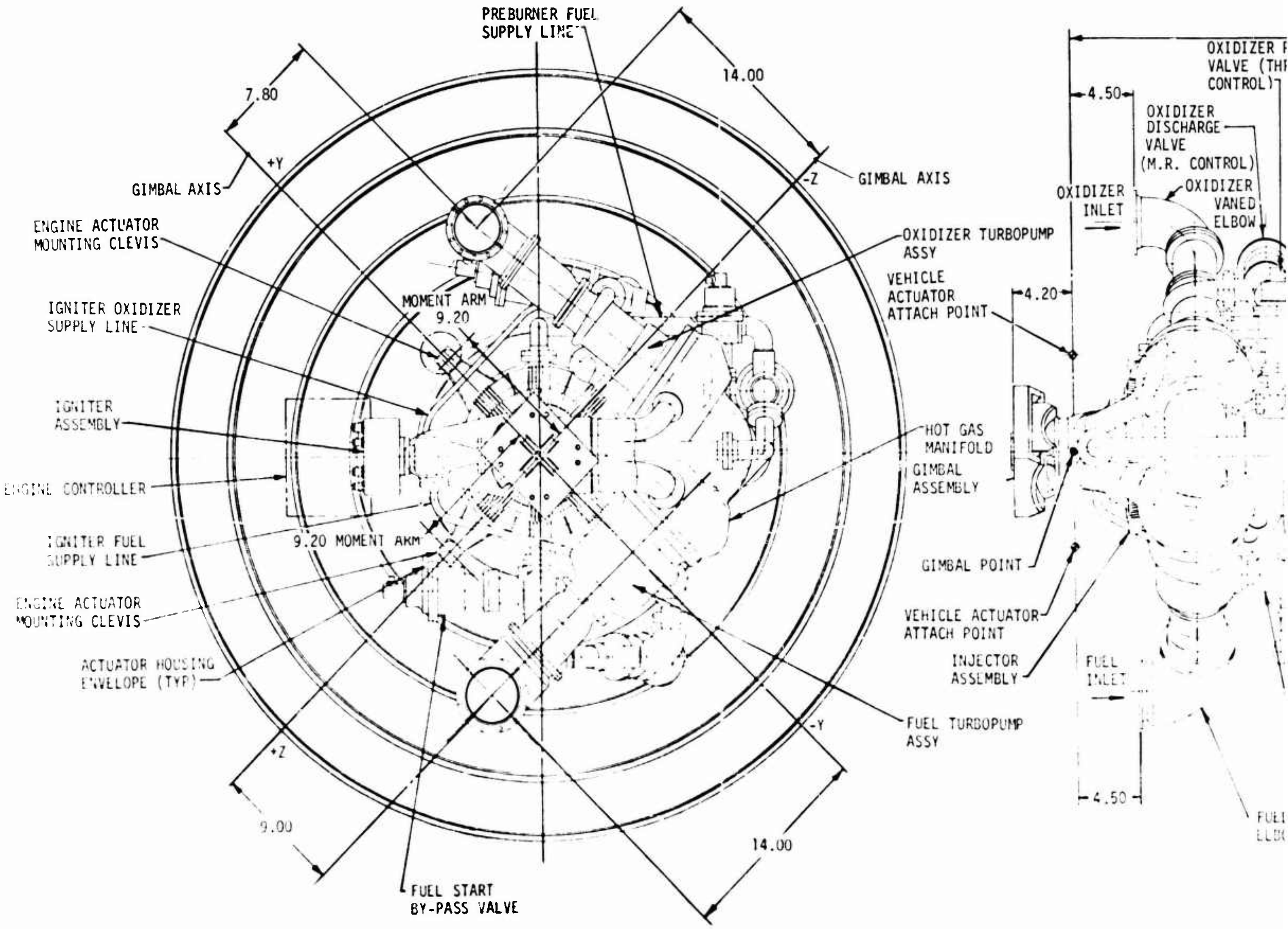
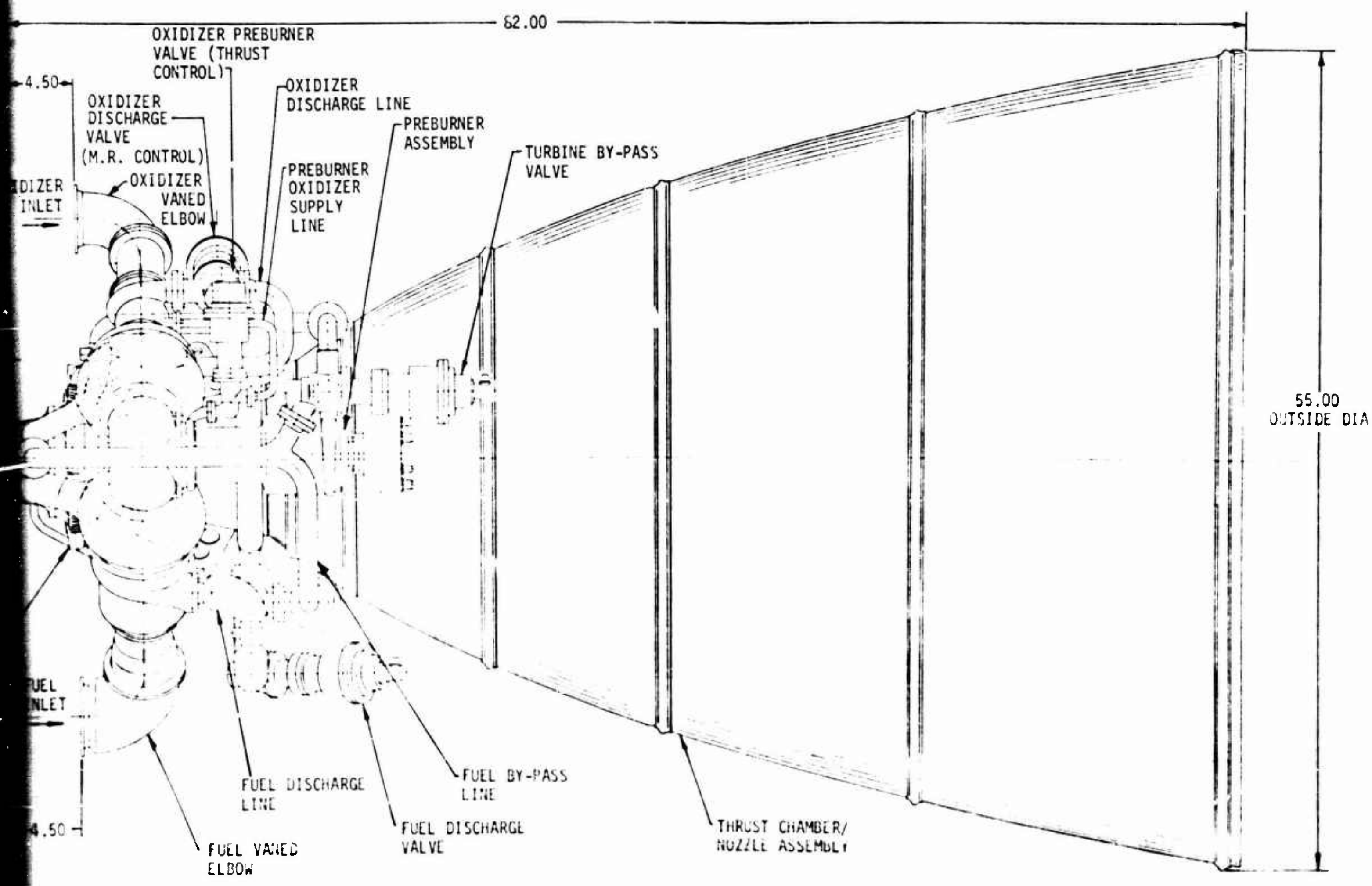


Figure 227. 25,000 lb Thrust Engine l



?rust Engine Layout

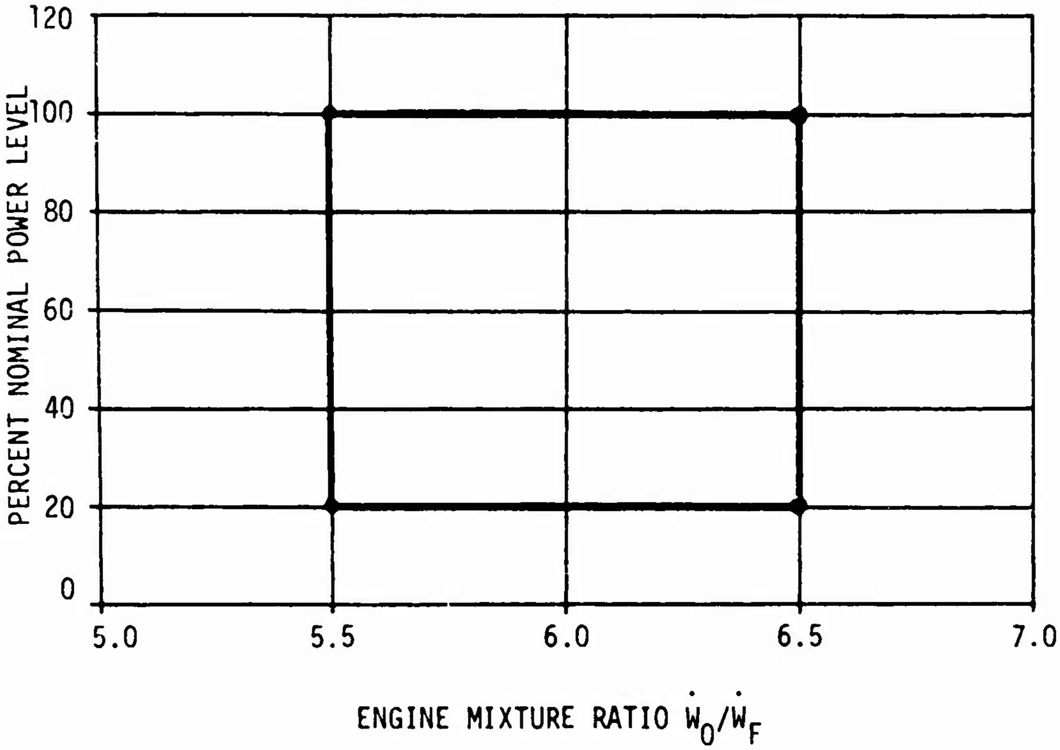


Figure 223. Engine Characteristics, Required Operating Envelope (Sheet 1 of 4)

PARAMETER	VALUE
Vacuum Thrust, Lbs.	25,000
Engine Mixture Ratio	6.0
Chamber Pressure, psia	1800
Nozzle Configuration	Min. RAO-Fixed All H_2 Regen. Cooled
Nozzle Exit Area Ratio, ε_o	290:1
Vacuum Specific Impulse, Sec.	465.7
Engine Weight, Lbs.	459.3
Number of Vacuum Starts	60
Service Free Engine Firing Cycles	60
Service Life Between Overhauls-Thermal Cycles (Reusable Mode)	300
Total Thermal Cycles (4 overhauls)	1500
Service Free Engine Run Time, Hours	2
Service Life between Overhauls, Hours	10
Total Service Life, Hours (4 Overhauls)	50
Maximum Single Run Duration, Sec.	1000
Maximum Time between Firings (Coast Time) Days	14
Minimum Time between Firings (Coast Time) Min.	10
Maximum Storage Time in Orbit (Dry) Weeks	52
Fuel Pump NPSH, Ft of H_2	60
Oxidizer Pump NPSH, Ft OF O_2	16
Gimbal Angle (Square Pattern) Degrees	5
Gimbal Acceleration, Rad/Sec^2	5
Min. Natural Frequency of Gimbal System	10

Figure [illegible]. Engine Characteristics, Required Operating Envelope (Sheet 2 of 4)

PARAMETER	VALUE
THROTTLING (VACUUM THRUST) RANGE TOLERANCE MAX CHANGE COMMAND RATE MAX CHANGE RATE	 Continuously variable 5.0:1 $\pm$ 3% of commanded value Step $\leq$ 320 lb/.010 sec
MIXTURE RATIO RANGE	 5.5 - 6.5 @ 5.0:1 Throttling
VACUUM I_S, SEC 5:1 THROTTLING 5.5 - 6.5 MIXTURE RATIO	

Figure [illegible]. Engine Characteristics, Required Operating Envelope (Sheet 3 of 4)

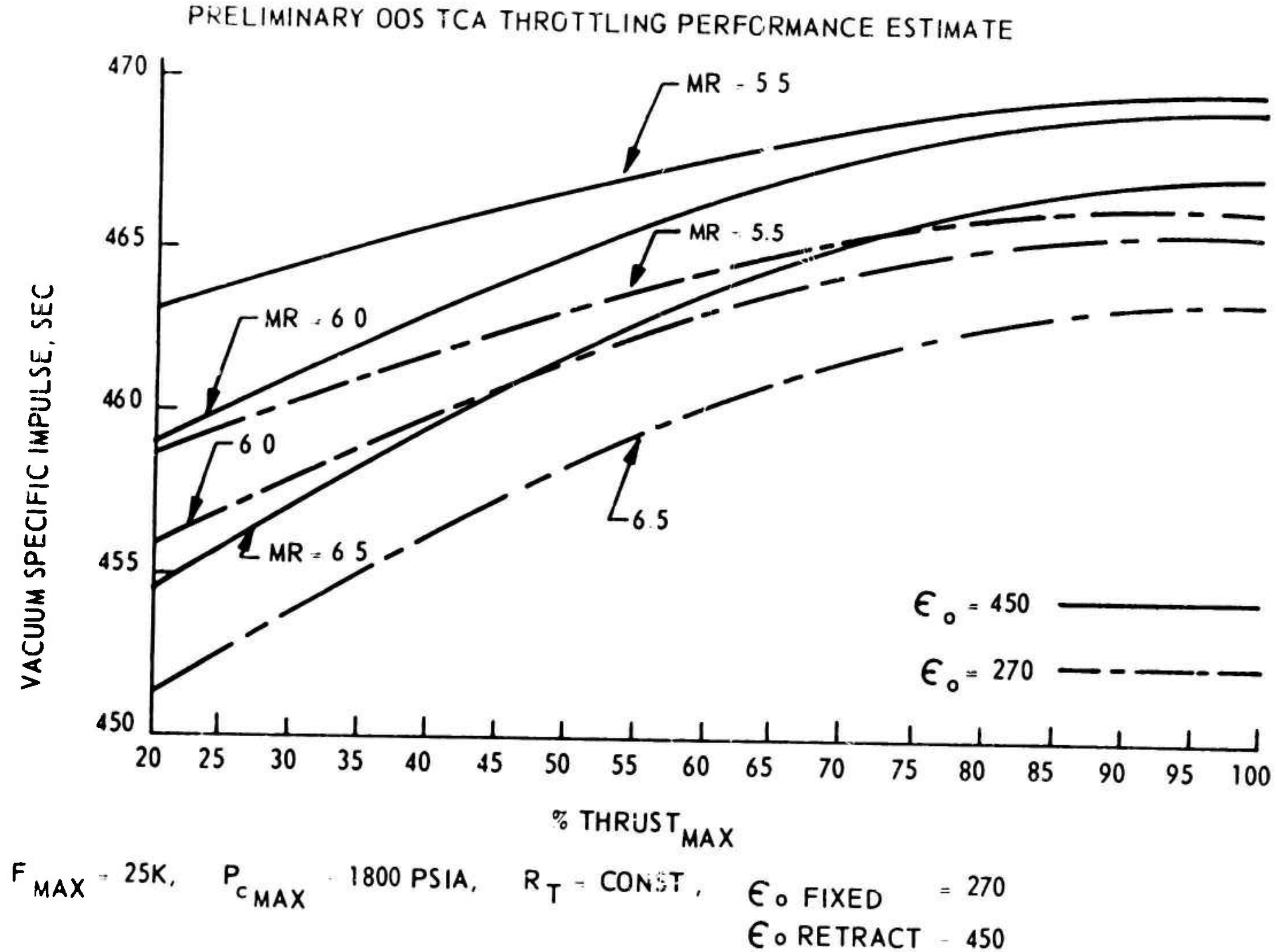
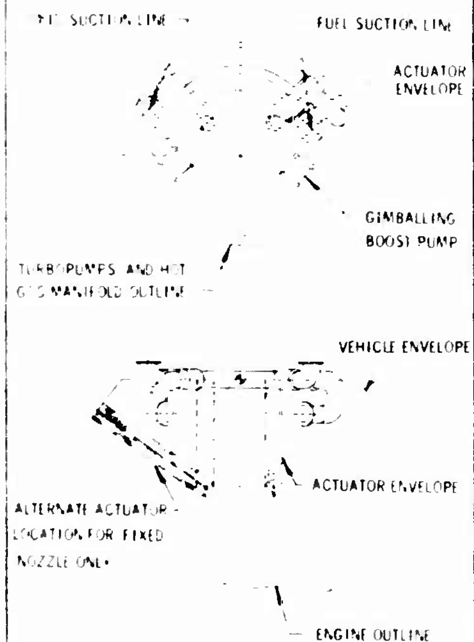
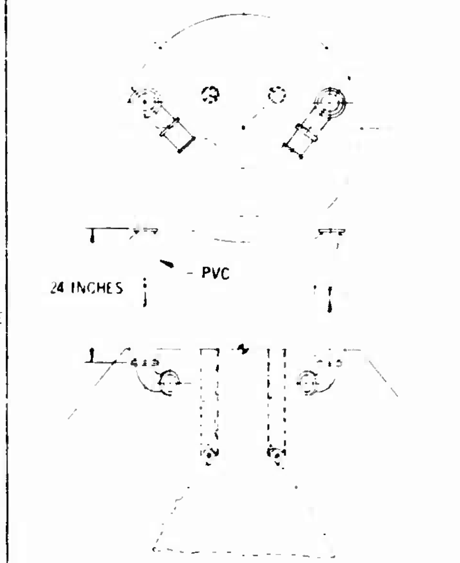
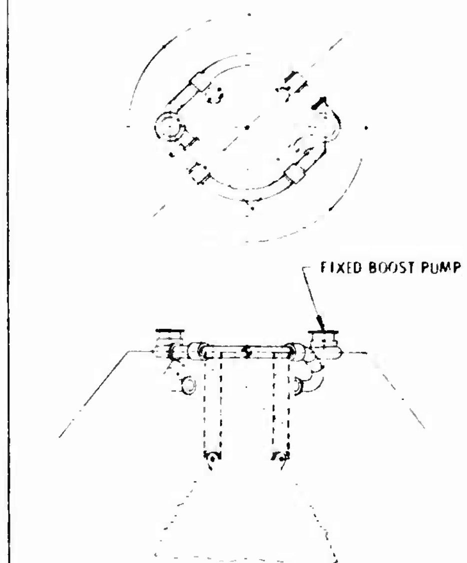


Figure 228. Engine Characteristics, Required Operating Envelope (Sheet 4 of 4)

TASK I

ODS SUCTION LINE CONFIGURATION MATRIX

GIMBALLING BOOST PUMPS

INSTALLATION CONFIGURATIONS	OPTION 2	OPTION 1	OPTION 3
SPECIFICATIONS: GIMBAL ANGLE ± 5 FUEL FLOWRATE - 7.61 LB SEC OXID FLOWRATE - 45.6 LB SEC			

DESCRIPTION	GIMBALLING SUCTION LINES		PVC SUCTION LINES		SUCTION LINES ON BOTH SIDES	
PROPELLANT	FUEL	OXID	FUEL	OXID	FUEL	OXID
LINE DIA, IN	[illegible]	3.0	3.0	3.0	2.00	2.00
VACUUM JACKET DIA, IN	4.0	4.0	4.0	4.0	3.00	3.00
LINE LENGTH, IN	44.0	57.5	33.00	34.00	43.5	38.5
LINE DRY WEIGHT, LB	10.99	13.20	14.20	14.37	8.16	7.64
TOTAL DRY WEIGHT, LB	24.19		28.57		15.80	
LINE WET WEIGHT, LB	[illegible]	29.55	14.36	17.27	8.50	12.62
TOTAL WET WEIGHT, LB	41.61		[illegible]		21.12	
NPSH INCREASE, FT	[illegible]	3.0[illegible]	1.06	2.29	0	0
R & D COST	$90[illegible]		$16000		$8000 - 3000 FOR B.P. DRIVE LINES	

Figure 229. Suction Line Configuration

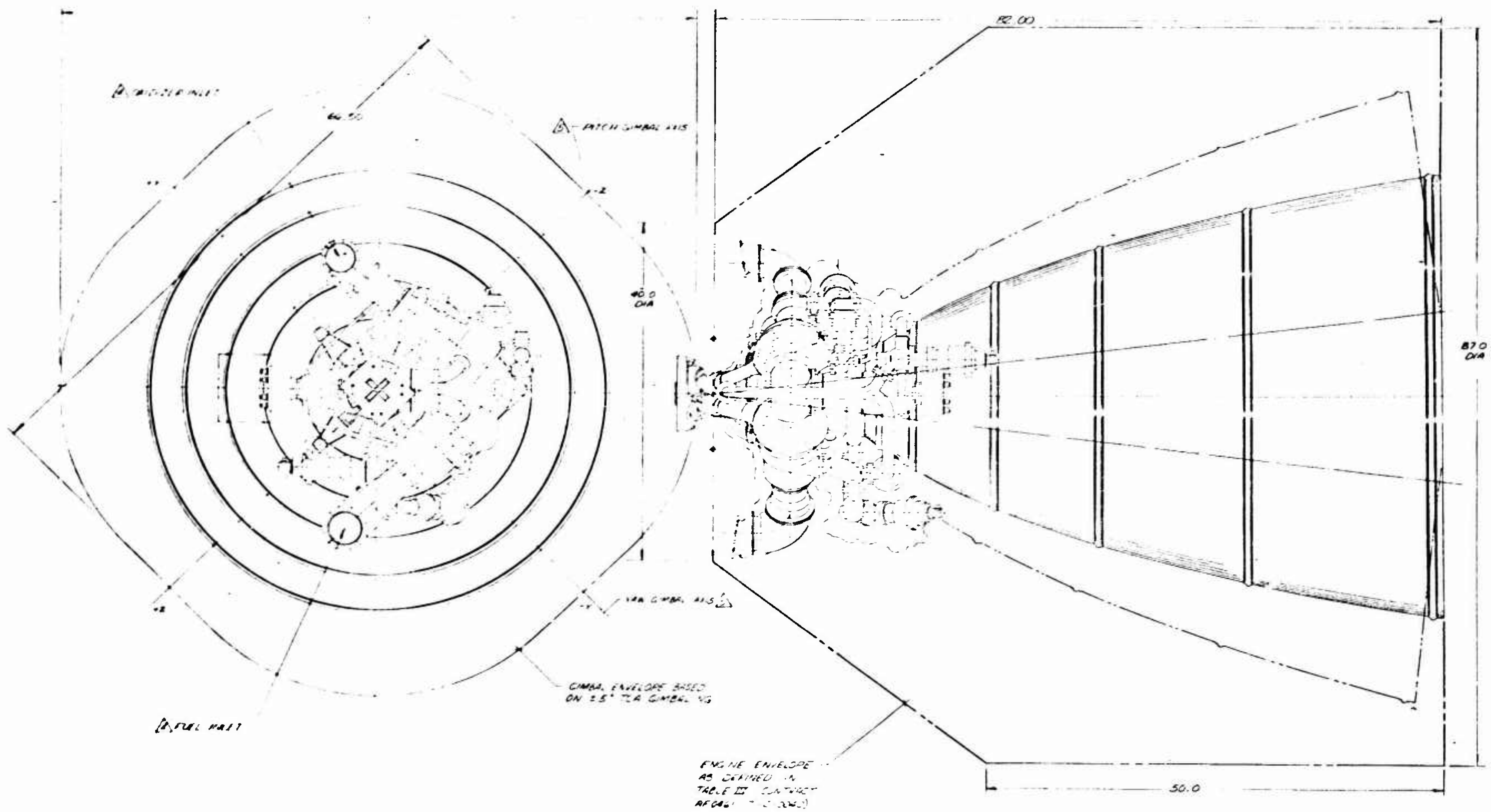


Figure 230. OOS Engine Gimbal Envelope Dimensions

TABLE LXIII

OOS ENGINE WEIGHT SUMMARY

Major Assembly	Weight, lb
Thrust Chamber Assembly (TCA) Consisting of: Injector Assy Thrust Chamber/Nozzle Assembly (Full H_2 Regen. Cooled to Exit $\epsilon = 290:1$) TCA Igniter Assembly	178.0
Preburner Assembly	24.7
Hot Gas Manifold	57.7
Turbopump Assemblies	67.5
Valves	43.1
Propellant Lines	20.6
Gimbal System (Gimbal Actuator assumed separate from engine system weight) Consisting of: Gimbal Assembly Gimbal actuator support structure	9.9
Engine Control System and Structure (Controller assumed separate from engine system weight) Consisting of: Harnesses Sensors Support Structure	45.5
Engine Attach Hardware and Seals	12.0
Engine Assembly Total Dry Weight	459.3

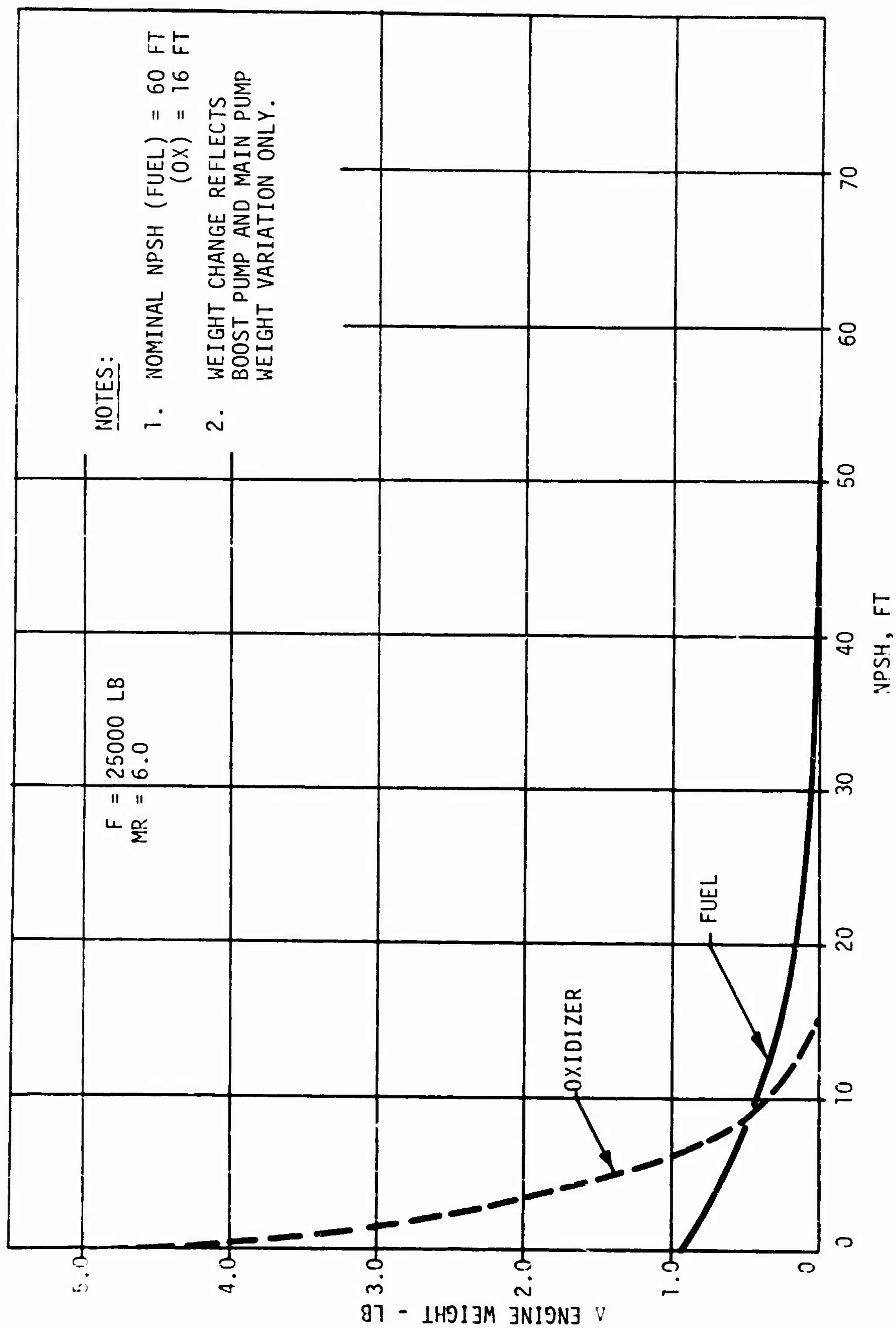


Figure 231. Engine Weight Change vs NPSH

Engine start and shutdown transient data are presented below and in Figure 232. Engine chilldown and idle mode operation data, including sequence of operation, are presented in Figure 233. Engine inert gas requirements are specified in Figure 234.

Engine Start Time	3.0 sec	Shutdown Time	1.5 sec
Start Impulse	18,780 lb/sec	Shutdown Impulse	9500 lb/sec
I_{sp} Avg.	440 sec	I_{sp} Avg.	400 sec
Propellant Consumed	46.82 lb	Propellant Consumed	23 lb
Thrust Rise Rate	37,000 lb/sec		

Alternate nozzle configurations for the OOS Engine, considering both fixed and retractable nozzle options, are shown in Figure 235. Engine weight and performance data with the nozzle options are shown in Figure 236 and the associated impact on vehicle payload is established and compared to the baseline fixed nozzle configuration.

The engine operates as follows: All of the hydrogen, after cooling the combustion chamber and nozzle, is combusted with a small amount of oxygen in the single preburner to produce turbine-drive gas. After passing through the turbines, this gas is exhausted into the thrust chamber where it is burned with the main oxidizer flow. Engine thrust is controlled by adjusting oxidizer flow in the single preburner. Throttling to 5:1 is achieved. Mixture ratio is controlled by controlling the main oxidizer flow to the injector.

b. Engine Purge Procedures

The engine purge system is shown schematically in Figure 237. Specific purge sequences are required for engine chilldown, preparation for start, start and shutdown. These sequences are discussed below. Engine helium requirements for a mission were determined to be 2.0 lb at 185 psia at the engine vehicle interface.

(1) Chilldown

During the chilldown of the oxidizer system P-2, P-13 and P-14, are energized to provide 25 psig He purge downstream of fuel discharge valve. The gas manifold He cooling flow is initiated by energizing P-12, P-7 and P-11. When the oxidizer TPA is chilled the oxidizer injector purges are initiated by opening P-3, P-4 and P-10 and the fuel purges are terminated by closing P-2, P-13 and P-14.

(2) Just Prior to Start

Before FS-1, the engine must be put in a ready-to-fire condition and checked for readiness. All the purge flows are switched to

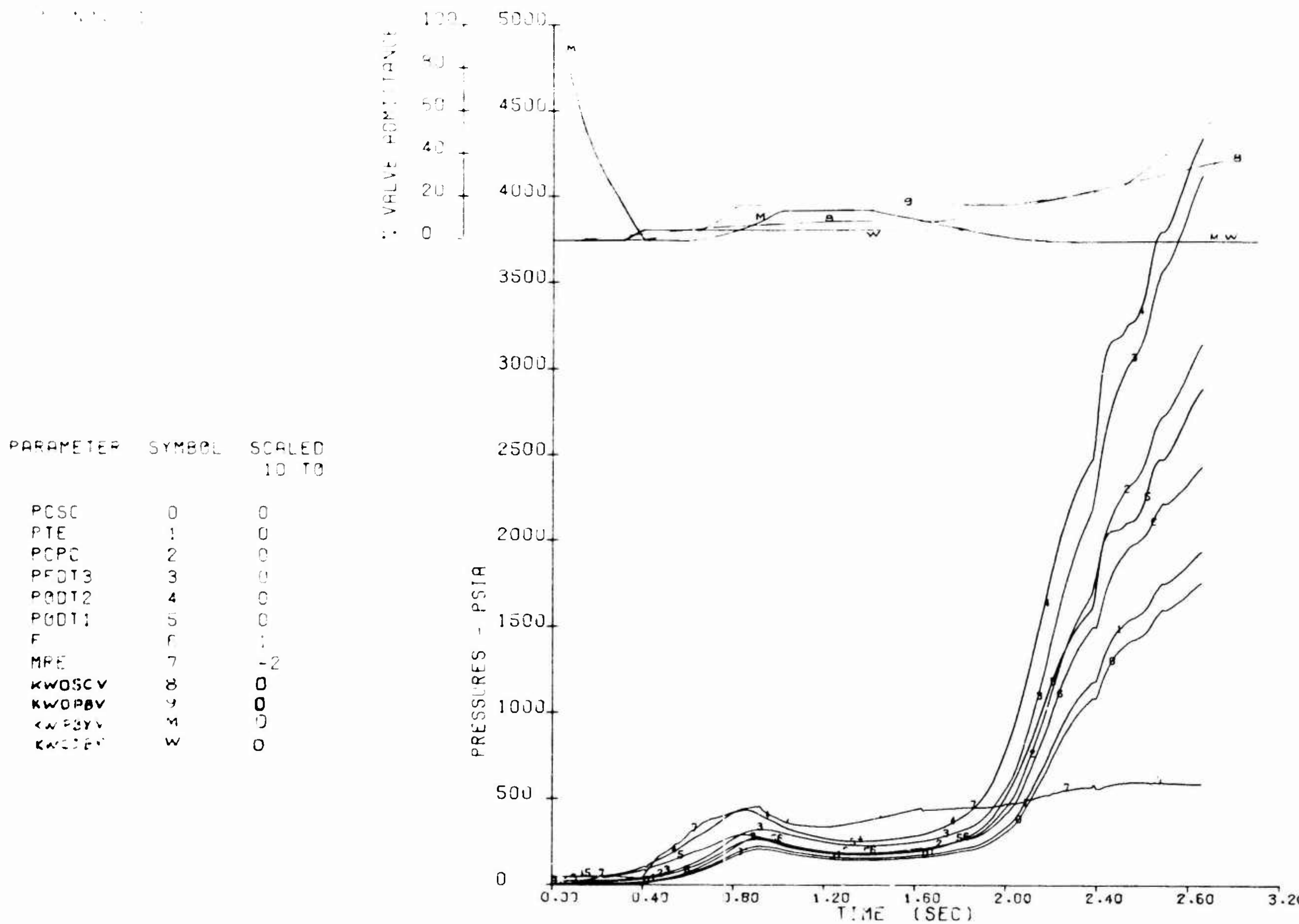


Figure 232. Engine Transient Data (Sheet 1 of 4)

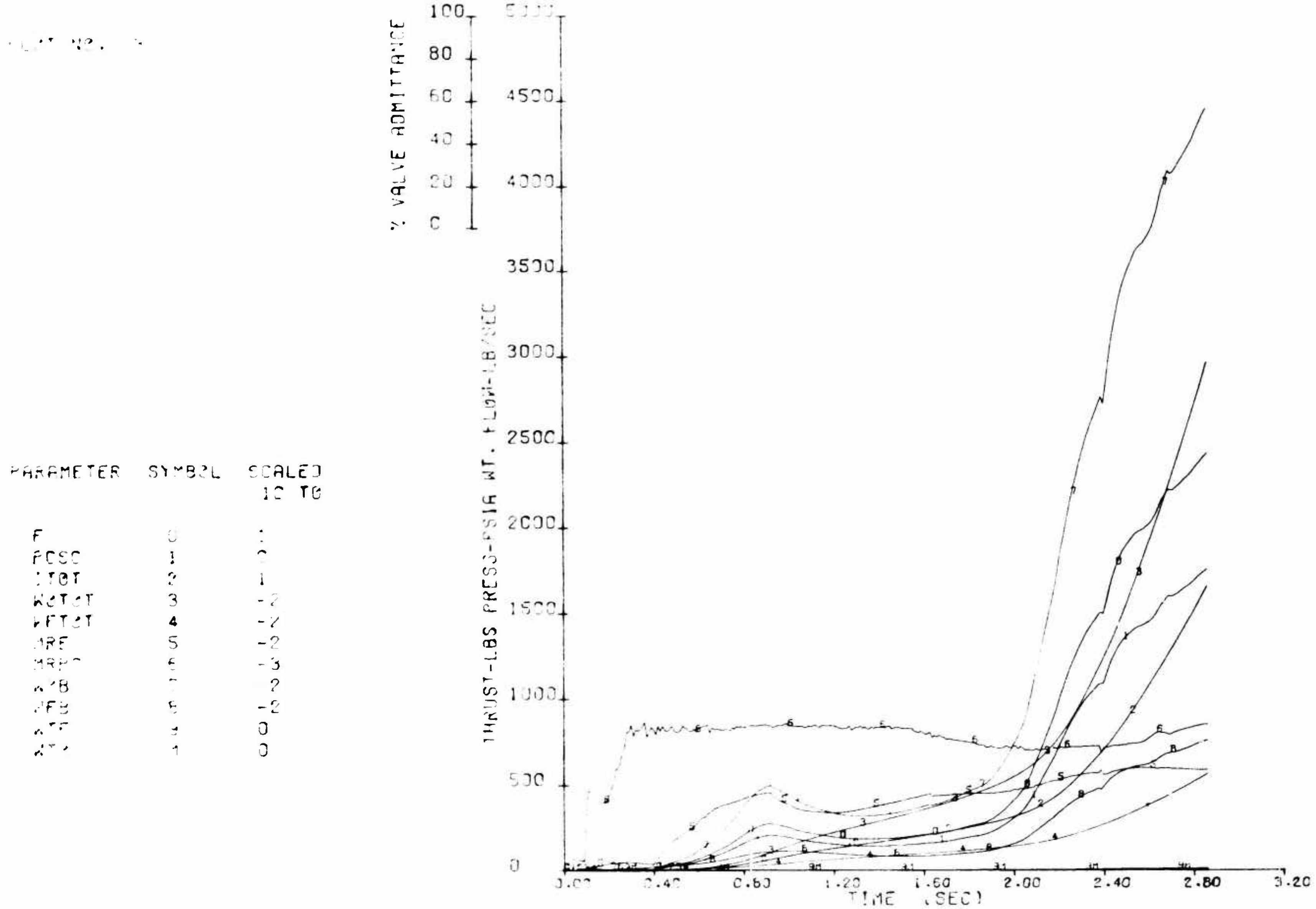


Figure 232. Engine Transient Data (Sheet 2 of 4)

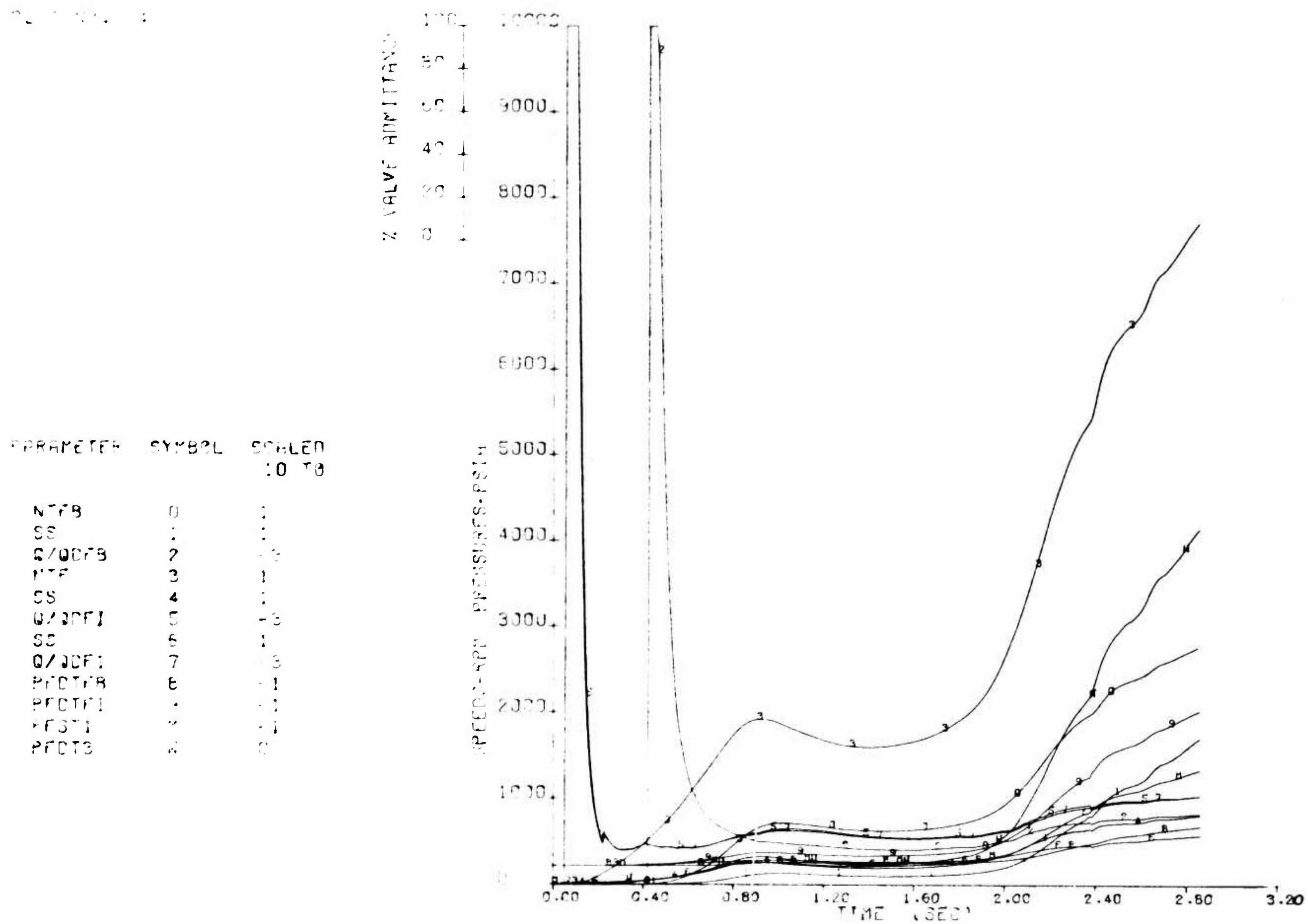


Figure 232. Engine Transient Data (Sheet 3 of 4)

PLOT NO.

PARAMETER	SYMBOL	SCALED 10 TO
NT0B	0	1
SS	1	1
Q/QD0B	2	-3
NT0	3	1
SS	4	1
Q/QD01	5	-3
SS	6	1
Q/QD01	7	-3
Q/QD02	8	-3
P0DT0B	9	-1
P3DT1	M	0
P3DT2	W	0

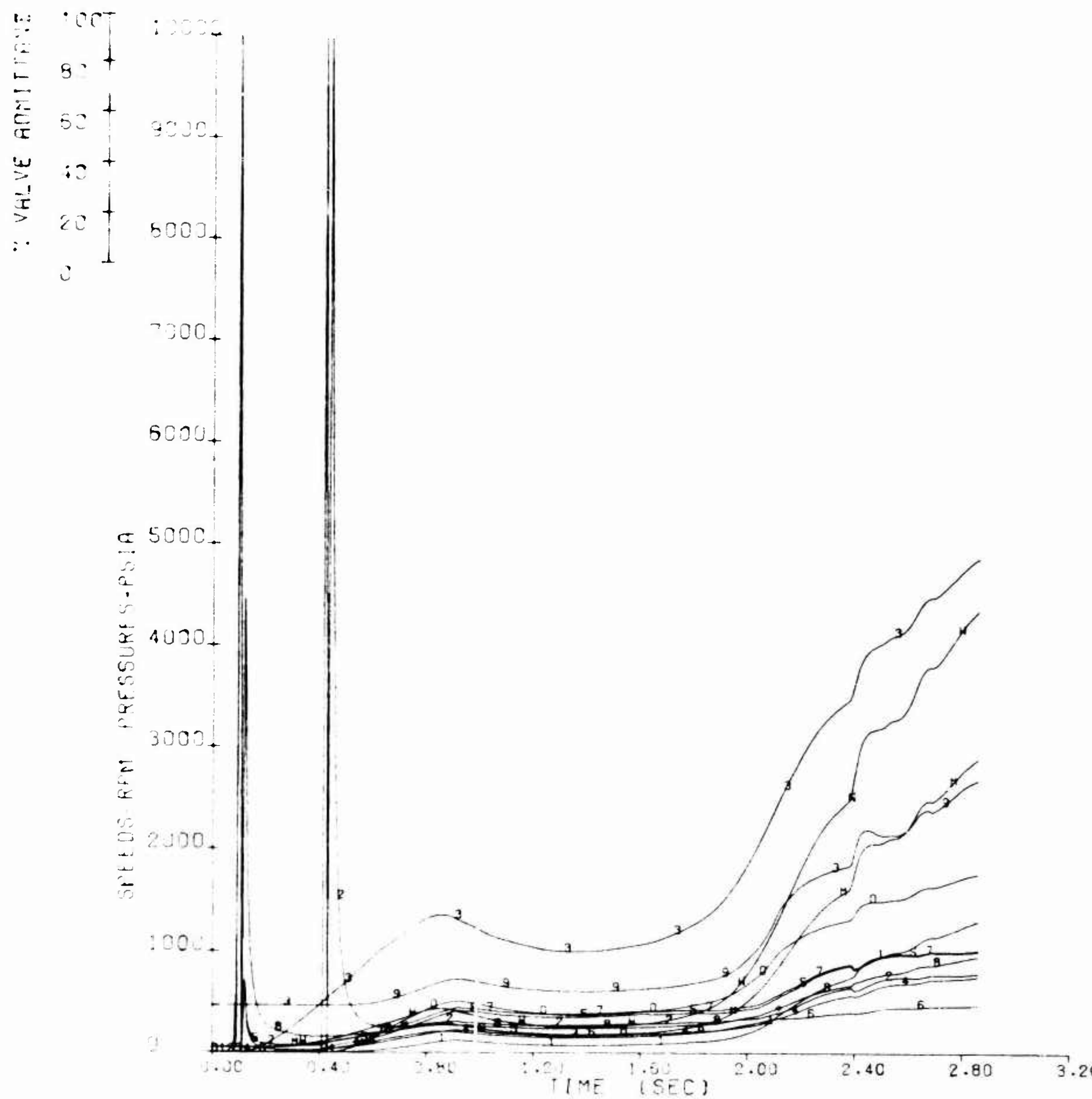
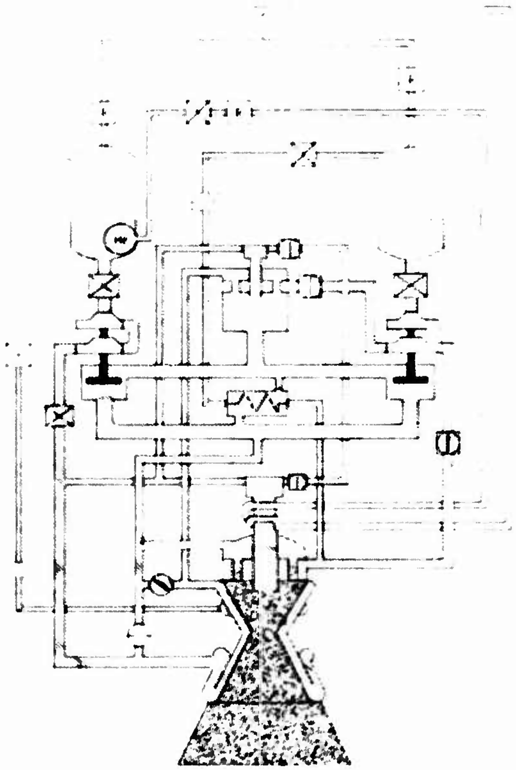
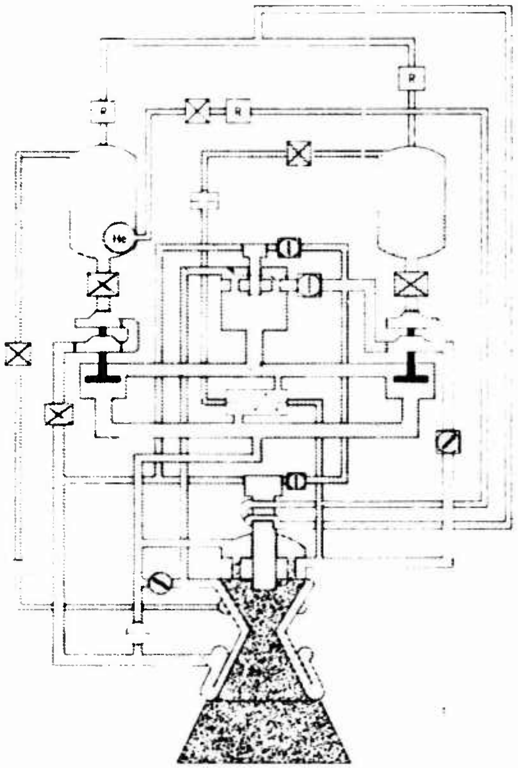


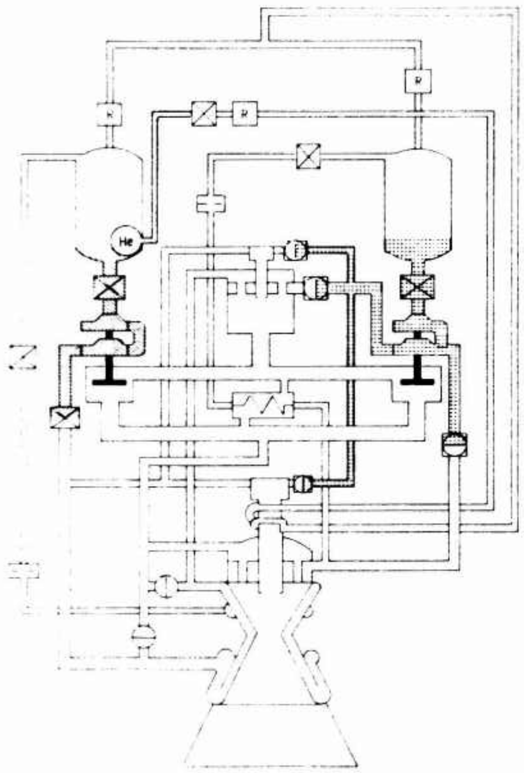
Figure 232. Engine Transient Data (Sheet 4 of 4)



- PREBURNER OXID VALVE OFF
- He TANK PRESSURIZATION ON
- PREBURNER BYPASS VALVE OPEN
- PREBURNER IGNITER VALVE OFF



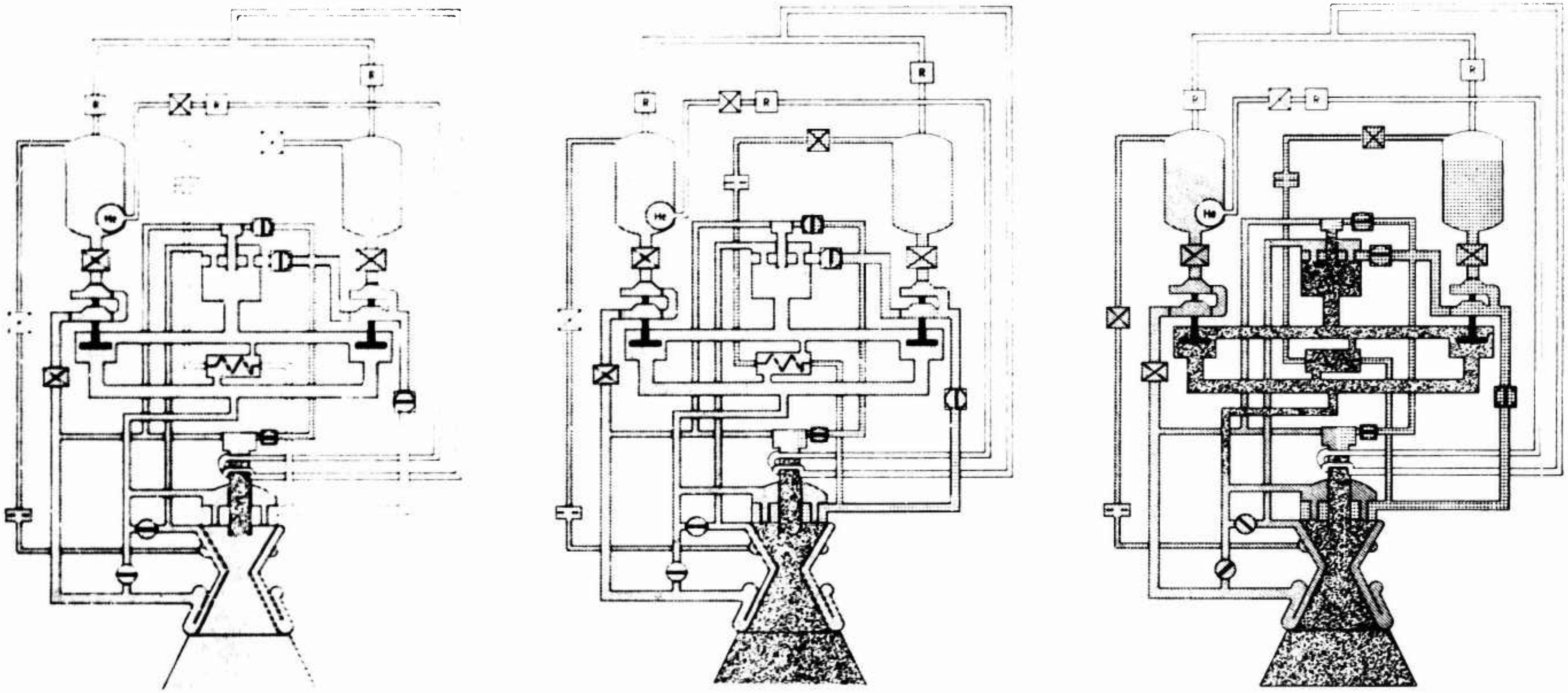
- OXID DISCHARGE VALVE MODULATE



- FUEL DISCHARGE VALVE CLOSED
- OXID DISCHARGE VALVE CLOSED
- He TANK PRESSURIZATION OFF

Figure 233. Idle Mode Operation (Sheet 1 of 2)

PRECHILLDOWN IDLE MODE PROPELLANT SETTLING | CHILLDOWN IDLE MODE (PRESSURE FED) | IDLE MODE TO ENGINE START



PRECHILLDOWN IDLE MODE PROPELLANT SETTLING

- He TANK PRESSURIZATION ON
- PREBURNER BYPASS VALVE OPEN
- FUEL DISCHARGE VALVE OPEN
- CHAMBER IGNITER ON

CHILLDOWN IDLE MODE (PRESSURE FED)

- OXIDIZER DISCHARGE VALVE OPEN
- IGNITER OXID VALVE MODULATED

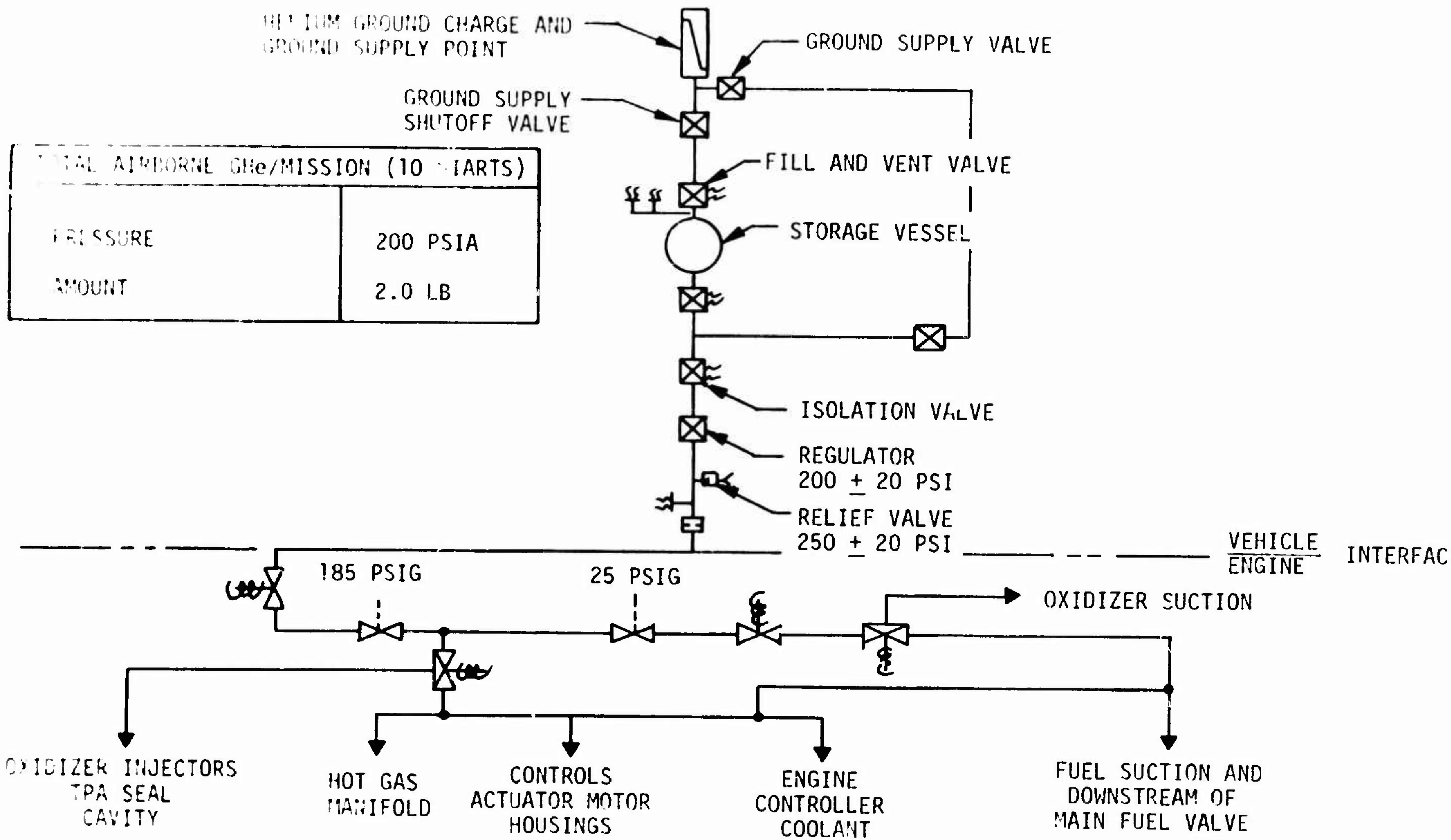
IDLE MODE TO ENGINE START

- PREBURNER BYPASS VALVE CLOSING
- He TANK PRESSURIZATION OFF
- CHAMBER BYPASS OPEN
- PREBURNER IGNITER ON
- PREBURNER OXID VALVE OPEN

Figure 233. Idle Mode Operation (Sheet 2 of 2)

• BLEED TYPE CHILLDOWN (Oxid. System Downstream of Discharge Valve)	PREVENT H_2 ENTERING THE OXID. MANIFOLD DURING PROPELLANT SETTLING & CHILLDOWN MODES
• START (Oxid. System Downstream of Discharge Valve)	CONTINUE PURGE TO PREVENT H_2 ENTERING THE OXID. MANIFOLD DURING H_2 LEAD AT START
• ENGINE OPERATION (Oxid. TPA Interpropellant Seal)	DILUTE OXID. & FUEL-RICH LEAKS INTO OXID. TPA INTERPROPELLANT SEAL CAVITY
• SHUTDOWN (Downstream of Oxid. Preburner & Discharge Valves)	FOR POSITIVE OXID. EXPULSION TO ENSURE CONTROLLED SHUTDOWN & SAFE OXID. EXPULSION
• CONTROL VALVE MOTOR PURGE	PREVENT SPARK INITIATED OXIDIZER IGNITION

Figure 234. Inert Gas Requirements (Sheet 1 of 2)



TOTAL AIRBORNE GHe/MISSION (10 STARTS)	
PRESSURE	200 PSIA
AMOUNT	2.0 LB

Figure 234. Inert Gas Requirements (Sheet 2 of 2)

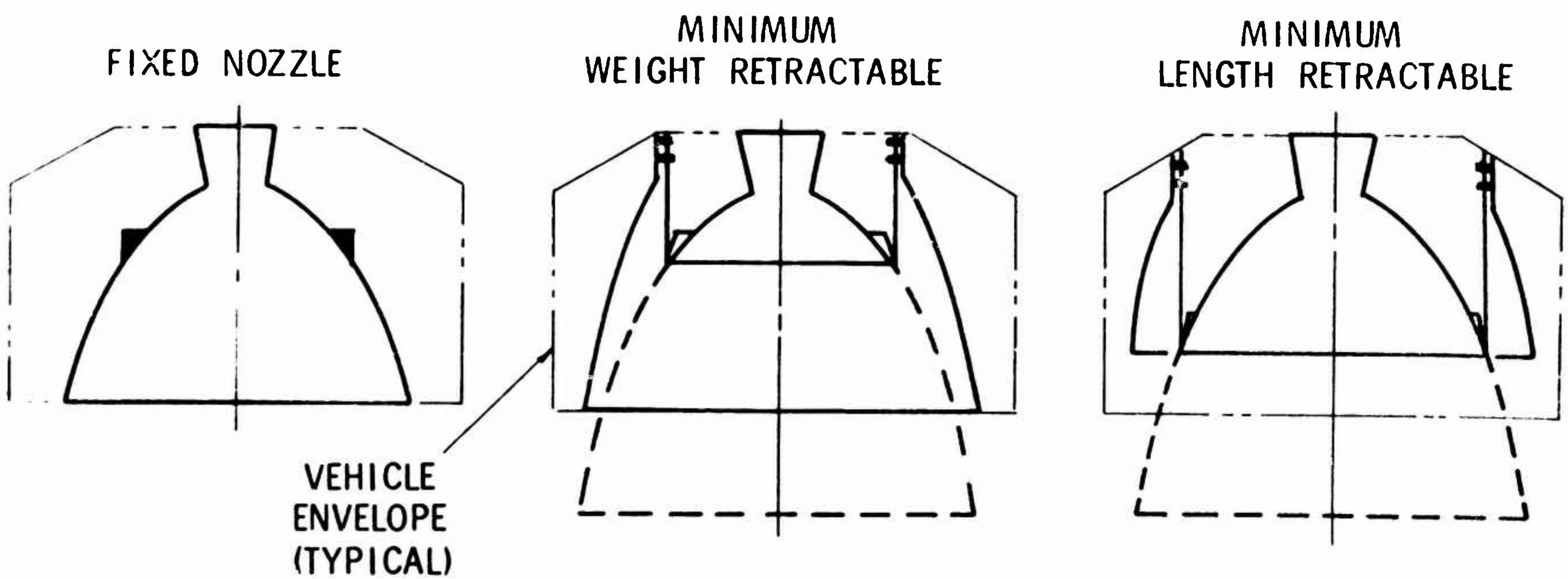


Figure 235. Nozzle Configuration

CYCLE: STAGED COMBUSTION

TURBINES: PARALLEL - SPLIT SHAFT (OTPA AND FTPA)

PERFORMANCE	NOZZLE CONFIGURATION MIN. WEIGHT RETRACTED	FIXED NOZZLE	(ALTERNATE) MIN. LENGTH RETRACTED
F, (LB)	25,000	25,000	25,000
MR	6.0	6.0	6.0
I_s (SEC)	468.2	465.7	468.2
P_c, (PSIA)	1800	1800	1800
F/P_c	13.88	13.88	13.88
R_t (IN.)	1.51	1.51	1.51
ENVELOPE			
ϵ_o	450	290	450
ϵ_{TRANS}	125	N/A	212
ENGINE OVERALL LENGTH (IN.) (L' = 6 IN.) (L' THROAT - GIMBAL = 13.5 IN.)	105 19	82.0	105 19
ENGINE STOWED LENGTH (IN.)	82.0	N/A	57.13
ENGINE EXIT DIA (IN.)	64.05	51.43	64.05
ENGINE WEIGHT, W_{BO} (LB)	512.3	459.3	565.1
*PAYLOAD			
157 ΔI_s	392.5	-	392.5
3.68 ΔW_{BO}	195	-	389.3
ΔPL (LOSS) (LB)	+197.5	-	+3.[illegible]

*PAYLOAD COMPARISON TO BASELINE FIXED NOZZLE,
ALL H_2 REGENERATIVELY COOLED

Figure 236. Engine Design Summary

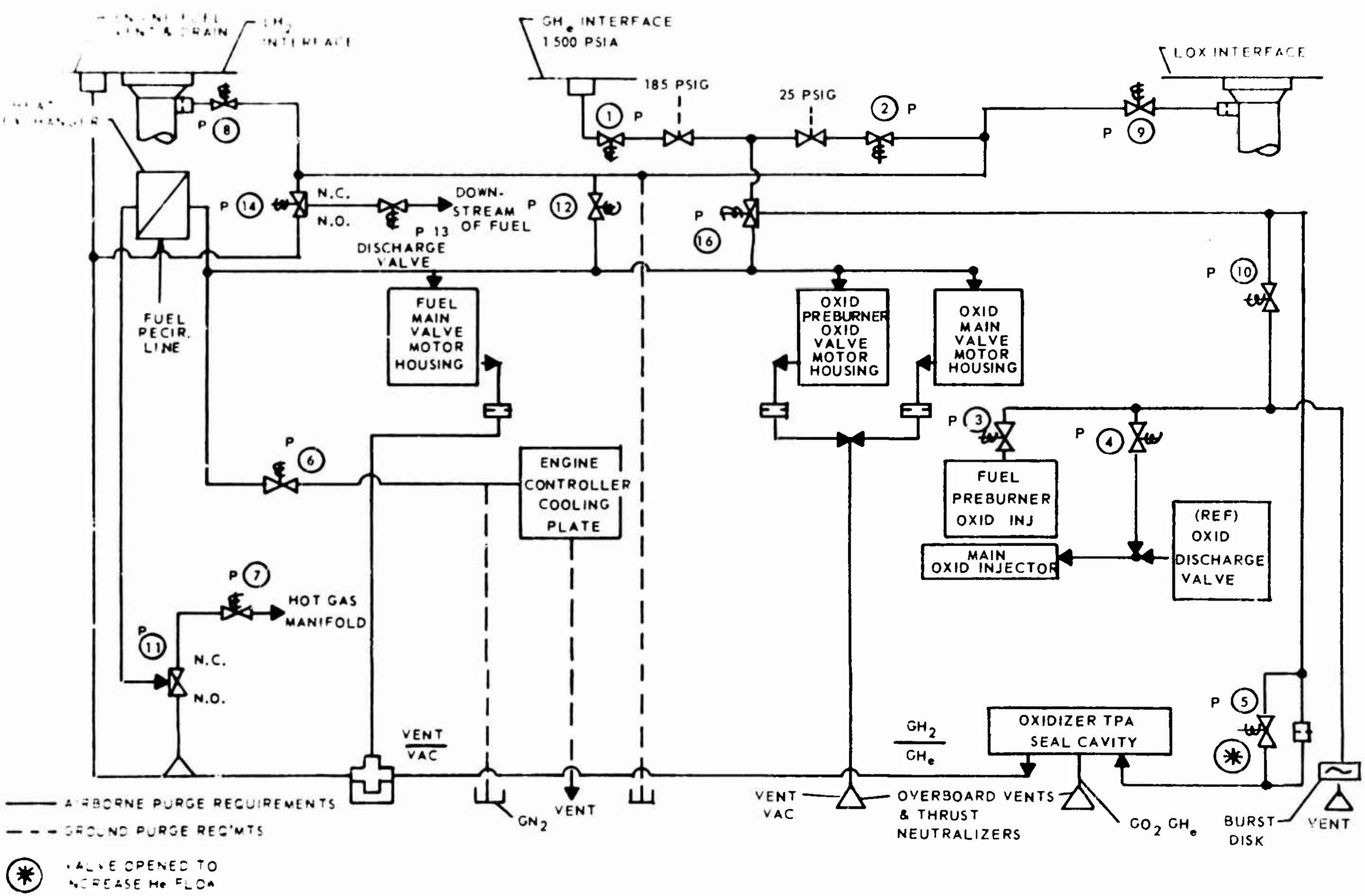


Figure 237. Engine Purge Schematic

25 psig helium and the gas manifold chill flow is terminated. The oxidizer injector purge pressure is increased to 185 psi by energizing P-15 to prevent fuel entering the oxidizer circuit during the fuel lead.

(3) Start

Oxidizer manifold purges are turned off by closing P-3, P-4 and P-10 at FS-1 + 0.250 seconds and the oxidizer TPA seal purge flow is reduced by closing P-5.

(4) Shutdown

At shutdown, the preburner oxidizer manifold is purged by opening P-3 and P-10. The thrust chamber oxidizer manifold is purged while the fuel discharge valve is closed, by opening P-3 and P-4. Immediately after shutdown, all purges are turned off by closing P-1, P-2, P-3, P-4, P-5 and P-10.

c. Engine Control Sequencing Requirement

Table LXIV describes engine sequencing requirements.

d. Propellant Settling

Propellant settling to provide approximately .1g is required to start the OOS Engine. Settling may be accomplished by either firing the APS or by bleeding fuel into the engine thrust chamber where it will expand to provide the required impulse. Detailed discussions on how the bleed system would operate is presented in Sections III,B,1,e, Engine Chilldown, and III,B,1,f, Engine Idle Mode.

TABLE LXIV

ENGINE CONTROL SEQUENCE

[illegible]	SEQUENCE OF EVENTS	OPERATION	CONTROL SYSTEM REQUIRED
[illegible]	1.Preburner bypass valve open 2.Cooling jacket bypass valve open. 3.Helium pressurization valve open. 4.Oxidizer manifold purge on. 5.Interpropellant seal purge on.	1.Causes fuel to bypass preburner so reduce pressure drop and prevent turbine rotation. 2.Causes fuel to bypass cooling jacket to prevent chocking flow. X3.Provides tank pressurization for pressure fed idle mode. 4.Prevents fuel entering oxidizer circuit. 5.Prevents interpropellant leak at oxidizer TPA.	(1 through 5) open loop control using valve position indicators including pressure transducers for purge system
Propellant Settling*	1.Fuel prevalve open. 2.Oxidizer prevalve open. 3.T.C.igniter spark plug energized 4.T.C.igniter ox valve open. 5.Fuel discharge valve open. 6.T.C.igniter MR closed loop control on. 7.T.C.igniter spark plug de-energized.	1.Allows fuel to flow into TPA up to fuel discharge valve. 2.Allows oxidizer to flow into TPA up to oxidizer discharge valve. 3.To ignite T.C. X4.Provides oxidizer flow to T.C.igniter. 5.Provides fuel flow T.C. igniter and T.C. - igniter operation is initiated and T.C. fuel flow expand nozzle providing settling impulse. 6.Once T.C.igniter is operating spark plug is de-energized.	(1,2,3,5 & 6) open loop control all valves except the T.C. Oxidizer igniter valve. Valve position indicators will be used. X (6) T.C.igniter oxid valve position requires closed loop control for MR because of low varying fuel flow during idle mode operation. MR will be determined by measuring fuel & oxidizer igniter inlet temp. & pressure.

*Assuming APS is not used.
X Required for Idle Mode Only

TABLE LXIV (cont.)

[illegible]	SEQUENCE OF EVENTS	OPERATION	CONTROL SYSTEM REQUIRED
Idle Mode	1.Oxidizer manifold H_e purge off. 2.Oxidizer discharge valve open. 3.Thrust chamber igniter pulsed. 4.Oxidizer discharge valve actuated to preset idle mode position.	1.Oxidizer H_e purge turned off to permit T.C. oxid. flow. 2.Allows oxidizer to flow to T.C. 3.Raises igniter MR momentarily to provide hot enough gases to ignite main chamber flow. X 4.After chamber ignition idle mode mixture ratio is controlled by repositioning oxidizer discharge valve.	(1 through 4) open loop control using valve position indicators and pressure sensors in purge system and thrust chamber.
Idle to Full Thrust	1.Energize preburner igniter spark plug. 2.Preburner igniter oxid. valve open. 3.Preburner bypass valve programmed closed. 4.Cooling jacket bypass valve programmed closed. 5.Preburner oxidizer valve opened. 6.Oxidizer discharge valve full opened. 7.Preburner igniter de-energized. 8.Preburner igniter oxid. closed. 9.T.C. igniter oxidizer valve closed. 10.Closed loop T.C.igniter control off. 11.Helium pressurization valve off. 12.Closed loop engine control on.	1.Provides energy source for preburner igniter. 2.Initiate preburner igniter operation. 3.Start fuel flow to preburner X 4.Start fuel flow through chamber cooling jacket. 5.Initiate preburner operation. 6.Ramp to full engine thrust. 7.& 8. Shut off preburner igniter after ignition is established. X 9.Preburner igniter oxidizer valve remains closed during engine operation. 10.Autogenous pressurization is used during eng. oper. 11.Engine operation, thrust & MR, are controlled by closed loop engine controller after start transient.	(1 through 10) The start transient from idle mode thrust to full thrust is time sequenced open loop controlled. Valve position and pump speeds and pressures are sensed to monitor transient. 12. During engine operation for either man or reusable systems a close loop control system is used. Expendable engines will use an open loop control system.

X Required for Idle Mode Only

TABLE LXIV (cont.)

[illegible]	SEQUENCE OF EVENTS	OPERATION	CONTROL SYSTEM REQUIRED
Shutdown Idle Mode	1. Closed loop control system off. 2. Preburner oxidizer valve closed. 3. Helium pressurization valve open. 4. Preburner bypass valve opened to idle mode position. 5. Cooling jacket bypass valve opened to idle mode position. 6. Oxidizer discharge valve closed to idle mode position.	1. Shutdown and idle mode are controlled by open loop system. 2. Preburner is shutoff during idle mode engine operates as pressure feed system. X 3. Provides tank pressure for pressure feed idle mode. X 4. Preburner is bypassed to prevent TPA rotation during pressure fed idle more. X 5. T.C. cooling jacket is partially bypassed to reduce pressure drop during pressure fed idle mode. X 6. Oxidizer valve is closed to preset position for idle mode to achieve desired TC MR.	X (1 through 6) Open loop control system is used during shutdown and shutdown idle mode. Valve positions, helium pressure, and chamber pressure are monitored during the idle mode.
Shutdown	1. Oxidizer discharge valve closed. 2. Oxidizer system purge on. 3. Fuel discharge valve closed. 4. Helium pressurization valve closed. 5. Prevalves closed.	1. Shutoff oxidizer flow to T.C. 2. Purges oxidizer system to ensure positive expulsion and controlled shutdown. 3. Shutoff fuel flow to engine system. X 4. Shutoff helium flow to propellant tanks. 5. Isolate propellant from engine system.	(1 through 5) Open loop control system is used for shutdown. Valve position, helium pressure and chamber pressure are used to monitor system.

X Required for Idle Mode Only

5. Impact on 25K Engine Design and Performance Resulting from Revised Operating Requirements

a. Summary of Requirements and Design Impact

The matrix which defines the varying design candidates which were to be analyzed is Table LXV. It is included for the convenience of the reader.

b. Number of Vacuum Starts

A vacuum engine start is differentiated from thermal cycle. Engine start is a transient operating condition contrary to a thermal cycle which goes through complete thermal equilibrium.

In the turbomachinery component, engine starts and thermal cycle are difficult to differentiate since the turbine thermal gradients are slow in developing and are dependent on burn duration and for this analysis are considered to be thermal cycle phenomenon.

Engine components effected by the number of starts are few and no engine configuration change will be induced by the increase or decrease of the number of engine start requirements from the 300 life requirement.

Turbomachinery components considered in the impact of changing starting requirements are the dynamic seals, bearings, and lift-off seals. The TPA will pass several critical speeds during the start transient resulting in considerable shaft movement. All dynamic seals are hydrostatic seals and are believed to be insensitive to radial shaft movement. The axial rolling contact bearings are preloaded and soft mounted and not sensitive to the number of engine starts. The lift-off seal is externally actuated prior to shaft rotation and is not expected to be sensitive to the number of engine starts.

The engine control valves are designed with hard seals and bellows to be insensitive to the number of actuations and space environment. A reduction in the requirement would not result in any change either in the actuators or valves.

The thrust chamber and gas generator systems have a capability far beyond the stated requirements and no design change would be made within the stated limits.

The thrust chamber dynamic loads during the start transient are minimized since no flow separation is expected at hard vacuum conditions.

TABLE LXV

MATRIX FOR 25,000-POUND THRUST ENGINE, VARYING DESIGN CONDITION ANALYSES

EFFECTS TO BE DETERMINED

NEW DESIGN CONDITIONS	A. Engine Specific Impulse	B. Engine Weight	C. Engine Life (Cycles & Hours)	D. Engine Envelope Dimensions	E. Engine Interface Dimensions	F. Engine NPSH Requirements	G. Engine Cooling Requirements	H. Engine Maintenance Requirements	I. Engine Complexity	J. Gimbal Power Requirements	K. Turbine Drive Cycle & TPA Operating Cndtns	L. Technology Improvements Needed	M. Difference in Demonstrator Engine Dev. Schedule & Costs	N. Difference in Engineering Dev. Schedule & Costs	O. Engine Start Transient	P. Nozzle Area Ratio	Q. Difference in First Production Unit Cost
1. 20 Vacuum Starts	X	X				X						X	X	X			X
2. 600 Vacuum Starts	X	X				X						X	X	X			X
3. 5.0 to 7.0 Nominal Mixture Ratio	X	X	X		X		X				X	X	X	X		X	X
4. 10 Thermal Cycles (Expendable)	X	X					X	X			X		X	X			X
5. 60 Thermal Cycles (Reusable)	X	X					X	X			X		X	X			X
6. 600 Thermal Cycles (Reusable)	X	X					X	X			X	X	X	X			X
7. 2-Hour Life (Reusable)	X	X					X	X			X		X	X			X
8. 20-Hour Life (Reusable)	X	X					X	X			X	X	X	X			X
9. 3° Gimbal Angle		X		X	X					X							X
10. 10° Gimbal Angle		X		X	X					X							X
11. 40 Rad/Sec^2 Gimbal Accel.		X	X		X			X		X							X
12. 100 to 400 Nozzle Area Ratio	X	X	X	X			X			X		X	X	X			X
13. Multi-Position Nozzle	X	X	X	X			X	X	X	X		X	X	X		X	X
14. 0 ft H_2 Pump NPSH		X	X		X				X			X	X	X	X		X
15. 15 to 60 ft H_2 Pump NPSH		X	X		X				X			X	X	X	X		X
16. 0 ft O_2 Pump NPSH		X	X		X				X			X	X	X	X		X
17. 2 to 16 ft O_2 Pump NPSH		X	X		X				X			X	X	X	X		X
18. 500 sec Maximum Run Time		X	X										X	X			X
19. 2000 sec Maximum Run Time		X	X										X	X			X
20. No Throttle Capability	X	X	X				X		X				X	X	X		X
21. 10:1 Throttle Capability	X	X	X				X		X			X	X	X	X		X
22. 2 Weeks Max Orbit Storage Time		X	[illegible]						X				X	X			X
23. 104 Weeks Max Orbit Storage Time		X	X						X				X	X			X
24. Idle Mode Operation	X	X	X		X	X			X		X	X			X		

c. Effect of Design Mixture Ratio Requirement

The change of engine design mixture ratio has a significant effect on all engine characteristics. The engine modifications required are dependent on the design objective. The approach taken was to achieve an engine life cycle capability as near as possible to that of the basic engine design.

(1) Coolant Requirements

The most severe case of chamber cooling is at MR = 5.0, since at this mixture the gas side heat transfer coefficient increases, resulting in higher ΔTw and Tw. The effects of mixture ratio on the chamber cooling requirements are shown in Figure 238 which is valid for a 290:1 thrust chamber nozzle area ratio and constant coolant jacket inlet pressure of 4150 psia. From Figure 239, the throat coolant Mach Numbers are selected for each engine design mixture ratio to match the thermal cycle capability of the basic engine design. To match the ΔTw at MR = 7.0 with the design ΔTw = 910°F, the Mach No. at MR = 7.0 has to be Ma = 0.45. At MR = 5.0 it is not possible to match the design ΔTw by a simple increase of the throat Mach No. since chamber coolant pressure drops would be prohibitive. Therefore, a maximum coolant Mach No. of Ma = 0.6 was selected as the limit for practical design which resulted in an increase of ΔTw and loss of thermal cycle capability.

The thermal cycle capability is presented in the following table for the selected throat Mach Number for all three mixture ratios indicating a slight loss at MR = 5.0.

LIFE CYCLE CAPABILITY VS MIXTURE RATIO

Mixture Ratio MR	5	6	7
Throat Mach No.	0.45	0.49	0.60
ΔTw, °F	920	870	870
Tw, °F	895	910	920
N_f, Life Cycle	940	1000	1000
% N_f Loss	6%	0	0

To achieve these engine life cycles, the coolant pressure drop at MR = 7.0 and MR = 5.0 have to be increased according to Figure 239. However, these pressure drop requirements are valid for constant pump discharge pressure and must be corrected for variable pump discharge pressure. In Figure 240 the coolant pressure drop characteristics as function of coolant inlet pressure is shown for three throat Mach Numbers Ma = 0.45, Ma = 0.49 and Ma = 0.60. Superimposed is the fuel pump characteristics which permit determination of the increased pump discharge requirements and actual coolant pressure drops for each design mixture ratio which are:

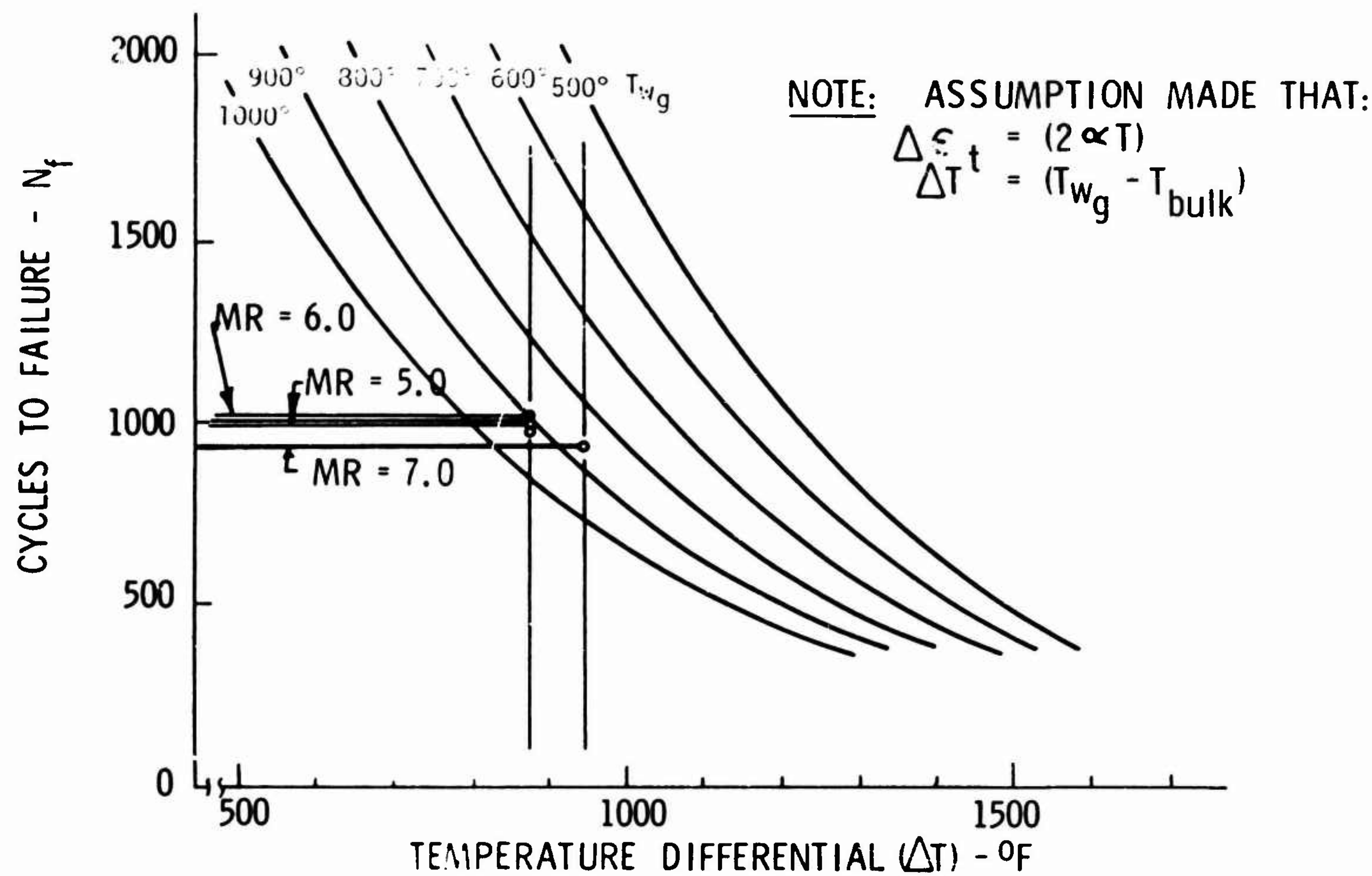


Figure 238. Zirconium Copper OOS Chamber LCF Requirements

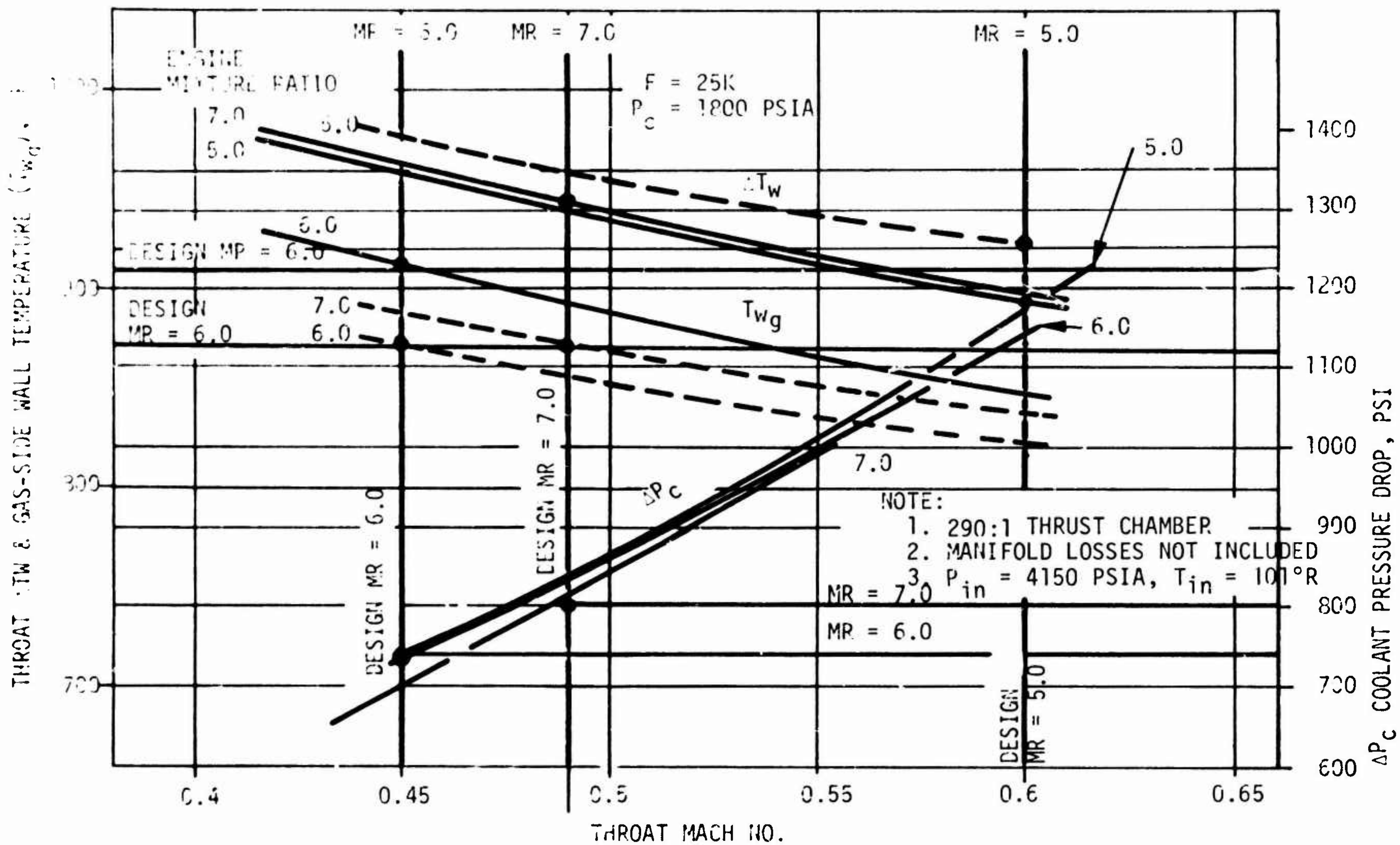


Figure 239. OOS Copper Nozzle Engine Mixture Ratio Study

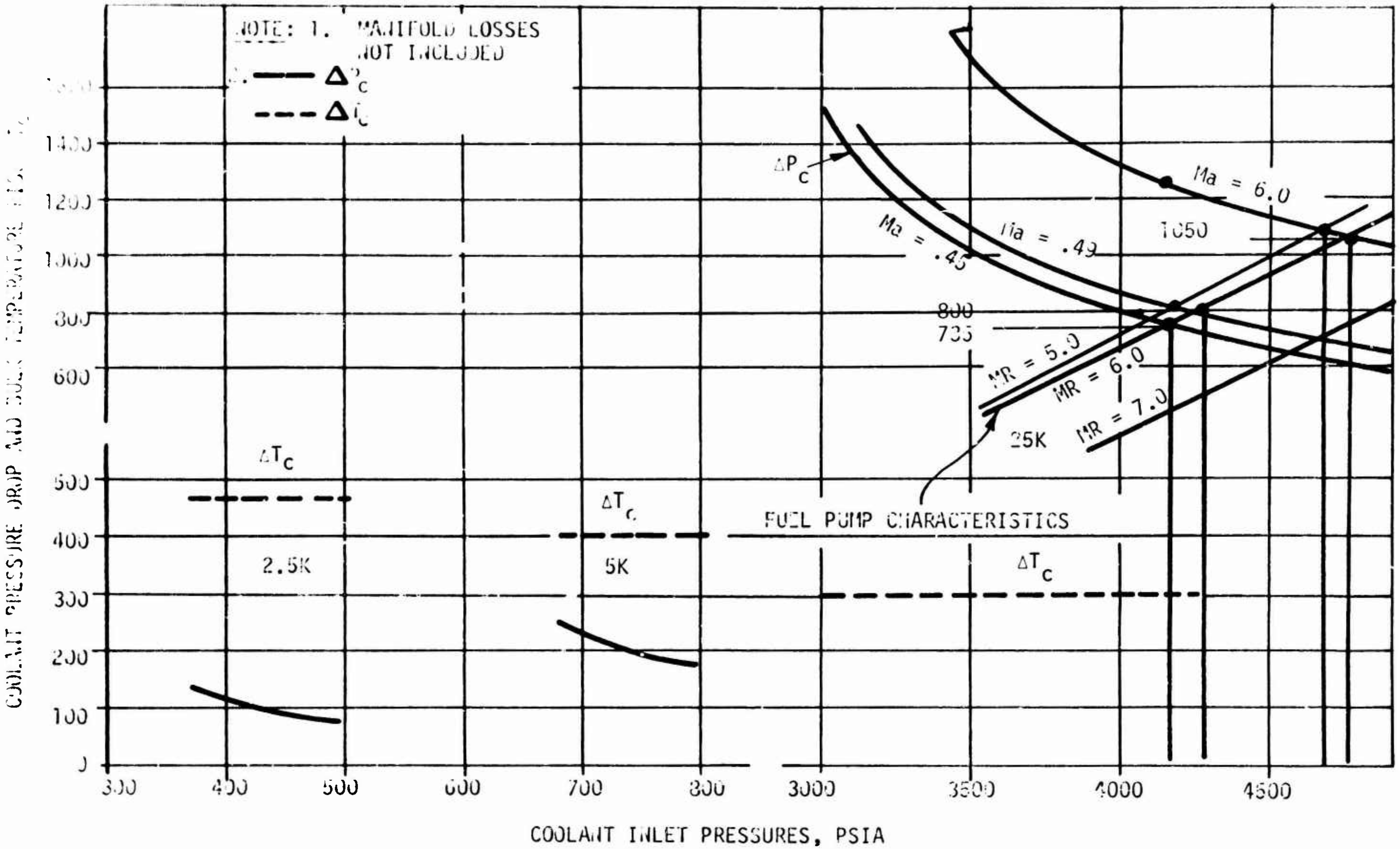


Figure 240. LH_2 Coolant Pressure Drop and Bulk Temperature Rise vs Coolant Inlet Pressure (Fuel Pump Discharge Requirements)

III, B, 5, Impact on 25K Engine Design and Performance Resulting from Revised Operating Requirements (cont.)

COOLANT AND PUMP DISCHARGE REQUIREMENTS
VS MIXTURE RATIO

Mixture Ratio	5	6	7
N_f Cycles	940	1000	100
Coolant Pressure Drop	1180	730	700
Coolant Bulk Temp Rise, °F	300	300	300
Pump Discharge Pressure, psia	4680	4170	4270
TPA Weight Change, lb	+10.8	0	+2.16

The change in design mixture ratio effects the turbopump requirements and also the weight of the turbopumps due to increased discharge pressure requirements.

The staged combustion cycle can achieve a power balance at all three design mixture ratios and no modification of the basic cycle is required.

(2) Engine Performance, Weight and Envelope Impact

The effect on engine performance and engine weight is strongly related to the nozzle area which can be accommodated within the fixed envelope. Based on the parametric study, the effects for an 82-in. stowed engine length are as follows: (6-in. combustion chamber length).

25K FIXED NOZZLE ALL REGEN COOLED

Mixture Ratio	5	6	7
Engine Length, in.	82	82	82
Max Diameter, in.	46.3	50	53.0
Nozzle Area Ratio	260	290	315
P_c, Chamber Pressure, psia	1800	1800	1800
I_s, Specific Impulse, sec	466.2	465.7	460.2
W_t Engine Weight, lb	474	459	454

(3) NPSH and Suction Line Diameter

The change of engine design mixture ratio has no effect on the engine NPSH capability since both the LO_2 pump and fuel pump have boost-pumps. The suction line diameters will, however, be slightly effected by the design mixture ratio. The values presented below are from the parametric engine design study.

Mixture Ratio	5	6	7
Thrust, lb	25K	25K	25K
LO_2 NPSH, ft	16	16	16
Fuel NPSH, ft	60	60	60
LO_2 Suction dia, in.	4.05	4.10	4.18
Fuel Suction dia, in.	3.80	3.60	3.46

d. Number of Thermal Cycles

The two components significantly effected by the thermal life cycle requirements are the thrust chamber and the turbine disk stresses.

The thrust chamber will reach its operating thermal equilibrium condition in a very short time and is relatively insensitive to burn time. The turbine disks experience a very slow temperature gradient and are, therefore, dependent on the burn duration.

The engine design chamber pressure selection is impacted by both the thrust chamber and turbine life cycle limitations, and a simultaneous solution can be established by an iterative process.

The turbine thermal cycle capability is dependent on the selected turbine temperature. To obtain high cycle life, low turbine temperatures are required, affecting the staged combustion cycle capability and chamber pressure selection. The relationship between thermal cycles and turbine temperatures is shown in Figure 241. indicating that the 600 thermal cycle requirements (1200 design cycles) requires a largely reduced turbine temperature of 1575°R as against 1860°R for the 300 cycle requirement. Since the cycle power balance is also dependent on the chamber coolant pressure drop requirement, the maximum obtainable chamber pressure is dependent on both turbine and chamber cycle life capability. Table LXVI power balance for 10, 60, 300 and 600 thermal cycles.

The thrust chamber life capability is shown in Figure 242 as a function of wall temperature and wall temperature gradient, and contains the design requirement for 60 cycles, 300 cycles and 600 cycles. To obtain the required chamber conditions, the coolant Mach No. have to be selected accordingly, resulting in the required coolant pressure drop requirements shown in Figure 242.

With the established coolant requirements, the recommended engine cycle balance vs cycle life requirements was analyzed and tabulated in the following table. The modified cycle balance is presented for the fixed nozzle concept only.

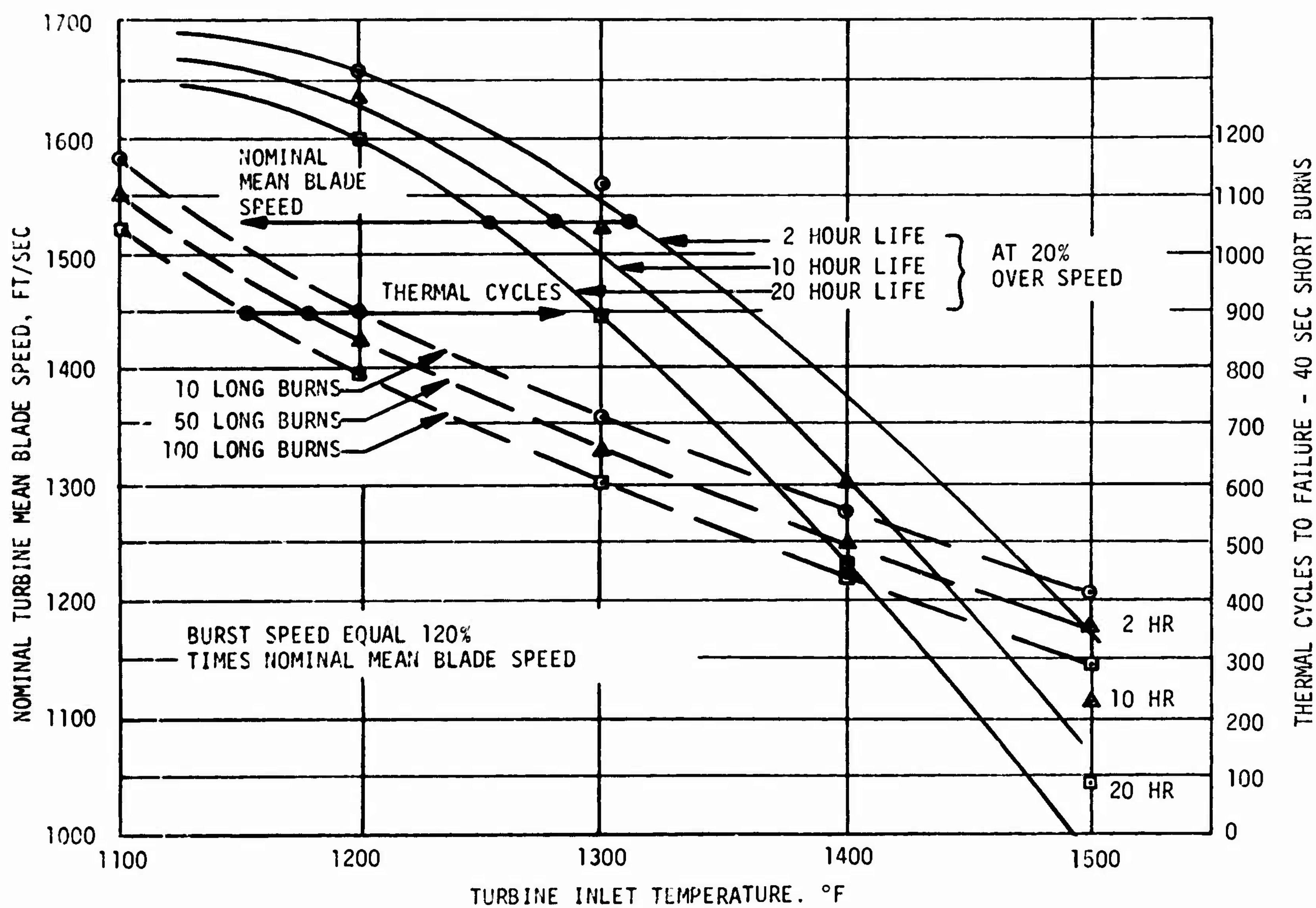


Figure 241. Turbine Disc Structural Criteria

TABLE LXVI

25K ENGINE TPA CYCLE LIFE SENSITIVITY

	10 Thermal Cycles		60 Thermal Cycles		300* Thermal Cycles		600 Thermal Cycles	
	Oxid	Fuel	Oxid	Fuel	Oxid	Fuel	Oxid	Fuel
Turbine Inlet Temp, °R	1947	1947	1947	1947	1860	1860	1575	1575
Turbine Mean Blade Speed, ft/sec	1100	1200	1100	1200	1100	1300	1100	1650
Thrust Chamber Pressure, psia	1900		1900		1800		1400	
Regen. ΔP, psi	550		550		735		1150	
Regen. ΔT, °R	310		310		295		250	
Area Ratio	283.3		283.3		290		215	
Delta Specific Impulse, sec	468.37 +2.7Δ		468.37 +2.7Δ		465.67 Ref.		463.37 -2.3Δ	
Pump Discharge Pressure, psia	3264**	4304	3264**	4304	3092**	4255	2405**	3789
Delta TPA Weight, Pound	+0.840Δ	-0.775Δ	+0.840Δ	-0.775Δ	Ref.	Ref.	-3.180Δ	-0.850Δ
Delta Boost Pump Weight, Pound	-0.017Δ	-0.014Δ	-0.017Δ	-0.014Δ	Ref.	Ref.	+0.0150	+0.013Δ
Delta TPA + Boost Weight, Pound	+0.823Δ	-0.789Δ	+0.823Δ	-0.789Δ	Ref.	Ref.	-3.165Δ	-0.837Δ
Suction Line Diameter, in.	3.13	3.60	3.13	3.60	3.13	3.60	3.14	3.61
Delta TPA + Boost Length, in.	+0.005Δ	-0.227Δ	+0.005Δ	-0.227Δ	Ref.	Ref.	-0.051Δ	+0.576Δ

Reference, Run 25 Aug 71 13:55:58.271

*Task I Baseline Turbopump
**First Stage Pump

F_{MAX} = 25K, MR = 6.0, STAGED COMBUSTION CYCLE
(NO HOLD TIME DATA)

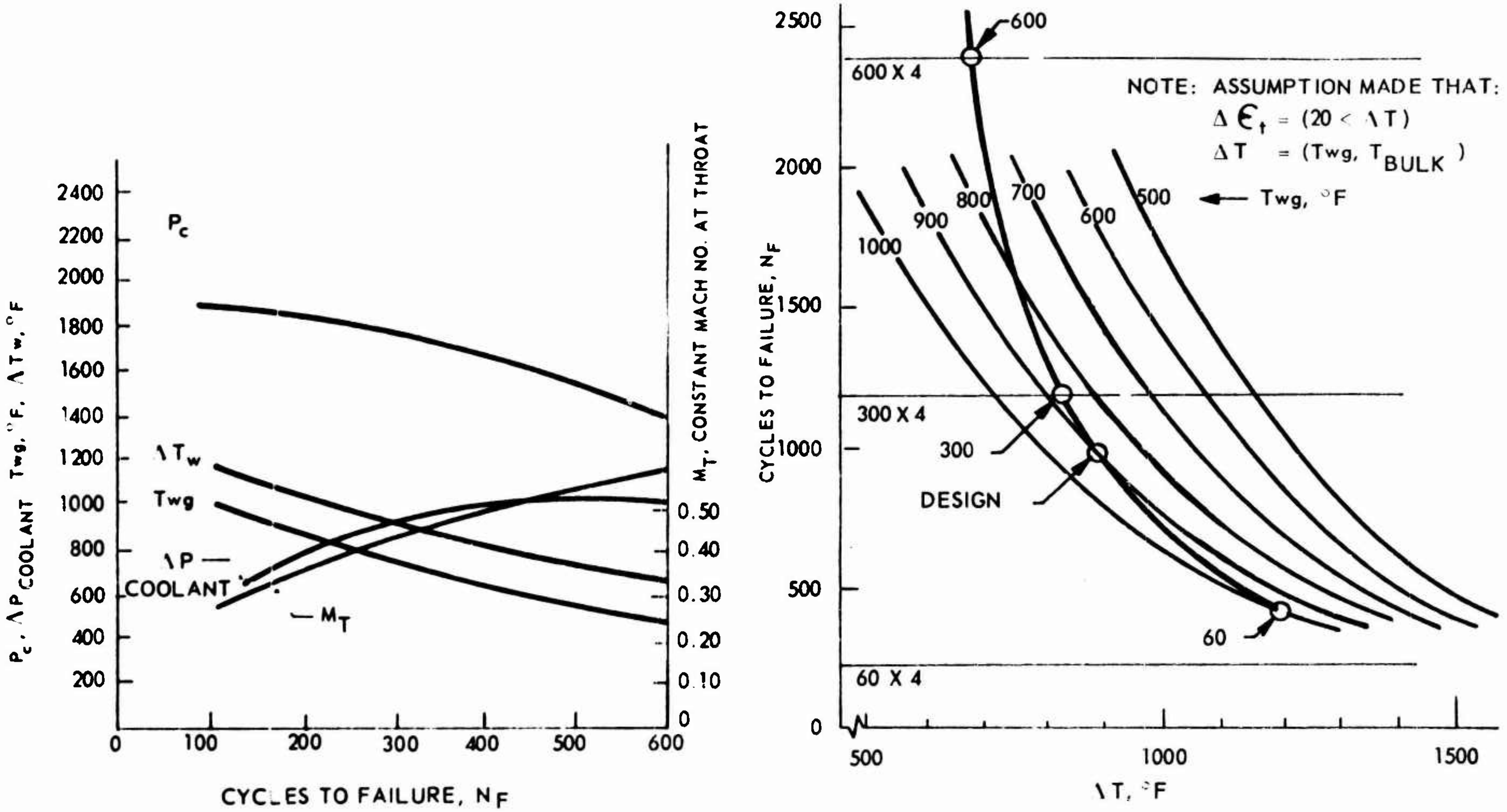


Figure 242. Chamber Life Cycle Requirements

FIXED ENGINE CHARACTERISTICS VS ENGINE CYCLE

No. Thermal Cycles	60	300	600
Thrust, lb	25K	25K	25K
Chamber Pressure, psia	1900	1800	1400
Area Ratio	300	290	230
Mixture Ratio	6	6	6
Specific Impulse, sec	466.30	465.6	463.8
Engine Weight, lb	375	380	401
F/P_c	13.16	13.90	17.85

Table LXVII shows the effect of requiring 10, 60, 300, and 600 thermal cycle capability per overhaul while keeping total duration fixed at 50 hours.

The baseline combustion components require no replacement over the entire 1500 cycle/50 hour service life, except for the main combustion chamber, which must be replaced at each overhaul (300 cycles). The combustion chamber copper liner has a 1220 thermal cycle capability when compared to minimum fatigue allowables, giving a factor of safety of 4.07.

Increasing the cyclic requirement to 600 cycles per overhaul (3000 total cycles) requires replacement of nearly all engine components, and the combustion chamber must be replaced after every other flight.

e. Engine Life Requirements

This section describes the impact of changing the engine life requirement from 10 hours to 20 hours and 2 hours of life respectively.

A structural analysis was made of the components covering:

- Turbopumps
- Main Chamber
- Main Injector
- Preburner
- Nozzle Extension

The results show only a minor impact on the turbomachinery design (Table LXVIII). In Figure 243 the relationship of turbine temperature, tip speed and life is presented. For a constant turbine temperature of 1860°R, the turbine tip speed can be varied to obtain the required turbine life. The

TABLE LXVII

THERMAL CYCLE CAPABILITY - TOTAL DURATION OF 50 HOURS

Component	Fatigue Life Capability	Expendible Mode 10 Cycles/O'Haul		Reduced Reqmnts 60 Cycles/O/Haul		Baseline Design 300 Cycles/O'Haul		Increased Reqmnts 600 Cycles/O'Haul	
		Replace Period	Factor of Safety	Replace Period	Factor of Safety	Replace Period	Factor of Safety	Replace Period	Factor of Safety
Main Combustion Chamber Zirconium Copper Liner T_{max} = 892°F	1220	None- 50 Cycles	24.4	None- 300 Cycles	4.07	O'Haul- 300 Cycles	4.07	Second Refurb. 240 Cycles	5.08
Regen-Cooled Nozzle Tubes Armco 22-13-5 Stainless Steel Tubing-707°F max	10,200	None 50 Cycles	High	None- 300 Cycles	34.00	None- 1500 Cycles	6.80	4th O'Haul 2400 Cycles	4.25
Main Injector Vanes-OFHC Copper Outer Platelets T_{max} = 932°F	9800 at 50 hr (1)	None- 50 Cycles	High	None- 300 Cycles	32.67	None- 1500 Cycles	6.53	4th O'Haul 2400 Cycles	4.08
Preburner U-Tube Liner Armco 22-13-5 Stainless T_{max} = 765°F	8800	None- 50 Cycles	High	None- 300 Cycles	29.33	None- 1500 Cycles	5.87	3rd O'Haul 1800 Cycles	4.88
Nozzle Skirt Extension AGCarb Material T_{max} = 2900°F	$>10^6$ (2)	None- 50 Cycles	High	None- 300 Cycles	High	None- 1500 Cycles	High	None- 3000 Cycles	High

(1) Only component affected by creep damage during extended duration operation
(2) Not sensitive to cyclic operation, total duration limited by ablation/erosion.

TABLE LXVIII

25K ENGINE TPA DURATION SENSITIVITY

NOTE: Assume Turbine Temperature = 1860°R	2 hr		Case I Baseline 10 hr		Reduce Shaft Axial Thrust 20 hr		Reduce Shaft Speed 20 hr	
	Oxid	Fuel	Oxid	Fuel	Oxid	Fuel	Oxid	Fuel
Thrust Chamber Pressure, psia	1800		1800		1800		1800	
Regen ΔP, psi	735		735		735		735	
Regen ΔT, °R	295		295		295		295	
Area Ratio	290		290		270		270	
Specific Impulse, sec	469.67		465.67		465.67		465.67	
Turbine Inlet Temp, °R	1860	1860	1860	1860	1860	1860	1860	1860
Turbine Mean Blade Speed, ft/sec	1100	1380	1100	1300	1100	1230	1100	1230
Pump Discharge Pressure, psia	3092	4220	3092	4255	3092	4315	3092	4476
Delta TPA Weight, lb	-0.03Δ	+0.800Δ	Ref	Ref	+0.020Δ	-0.200Δ	+0.843Δ	+14.628Δ
Delta Boost Pump Wt, lb	0. Δ	0. Δ	Ref	Ref	0. Δ	0. Δ	0. Δ	+0.279Δ
Delta Boost + TPA wt, lb	-0.03Δ	+0.800Δ	Ref	Ref	+0.20Δ	-0.2Δ	+0.843Δ	+14.907Δ
Delta TPA Length, in.	0. Δ	0.210Δ	Ref	Ref	0. Δ	-0.188Δ	0. Δ	+1.220Δ
Main TPA Shaft Speed, rpm	50,000	80,000	50,000	80,000	50,000	80,000	50,000	70,000
Thrust Brg Max Axial Force (includes preload), lb	120+	120+	78	75	53	20	53	65

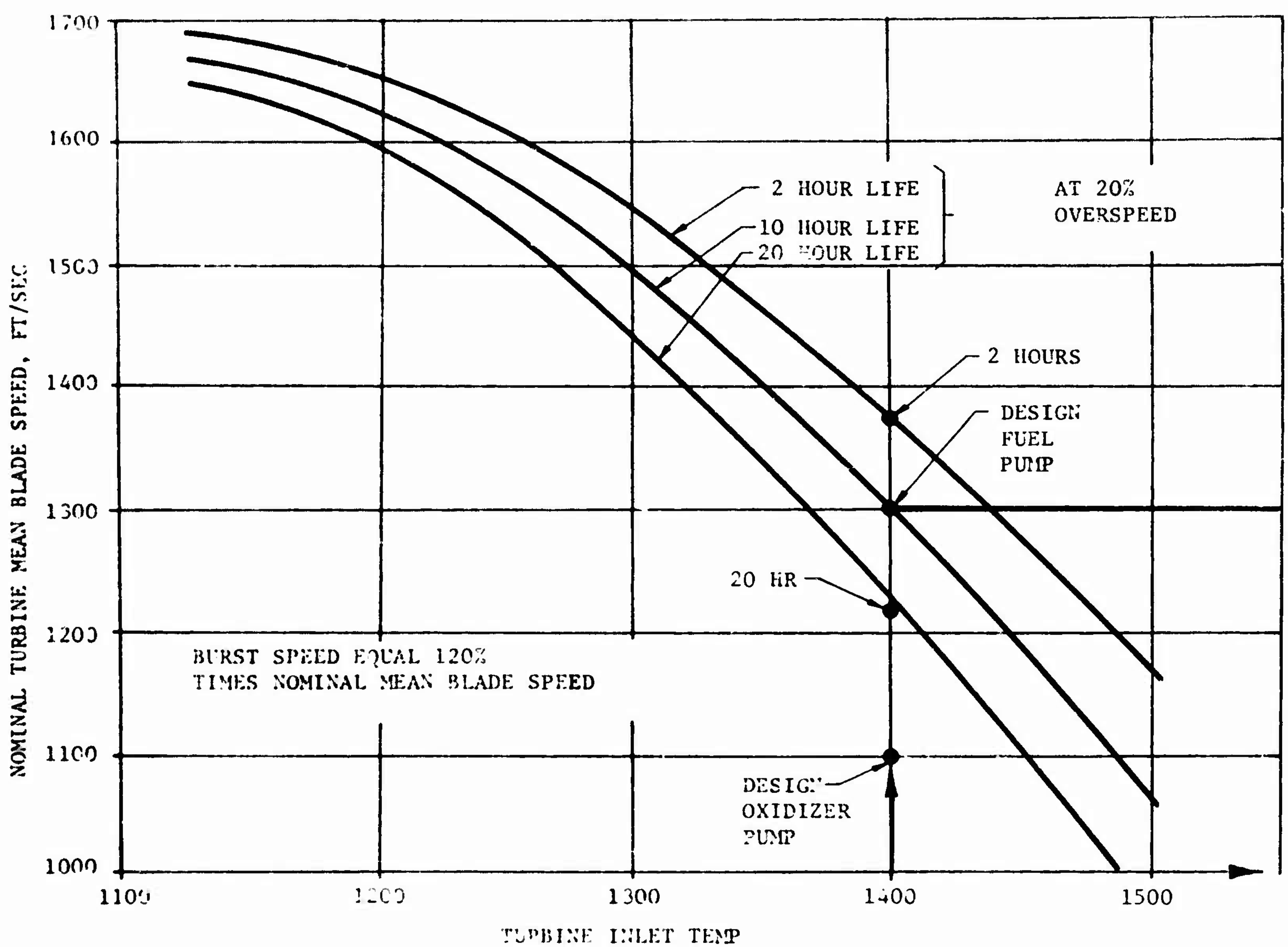


Figure 243. Turbine Operating Parameters

oxidizer design tip speed is 1100 ft/sec, meeting the 20 hours requirements without modification. The fuel turbine design tip speed is 1300 ft/sec and will be modified according to Figure 244 to meet the various life requirements.

The oxidizer turbine with a mean blade speed of 1100 ft/sec as compared to 1200 ft/sec for the fuel turbine will have a disk stress level that is 72% the magnitude of the fuel turbine stresses. This lower operating stress will increase the life capability of oxidizer turbine disk by a factor of 10 giving a life capability of 100 hours.

Changing of the fuel turbine tip speed effects the turbine efficiency which will require compensation by increased fuel pump discharge pressures. The impact of selected turbine temperature and tip speed on the fuel pump discharge requirements are shown in Figure 244 indicating very small effects. The impact of the shifting turbine design point is shown in Figure 245 indicating negligible TPA weight changes.

Total operating lifetime in the 10 to 100 hour range has little effect on OOS combustion component life or replacement costs. The low temperatures and thermal gradients necessary to meet the 6000 cycle service life (1500 x target safety factor of 4) keeps all materials selected except OFHC copper out of the creep damage range. The injector vanes still provide a 3.67 factor of safety at the maximum 100-hour service life investigated.

A summary of cycle capability for the required duration is shown in Table LXIX and Figure 246 indicating the chamber maintenance requirements against the component capability.

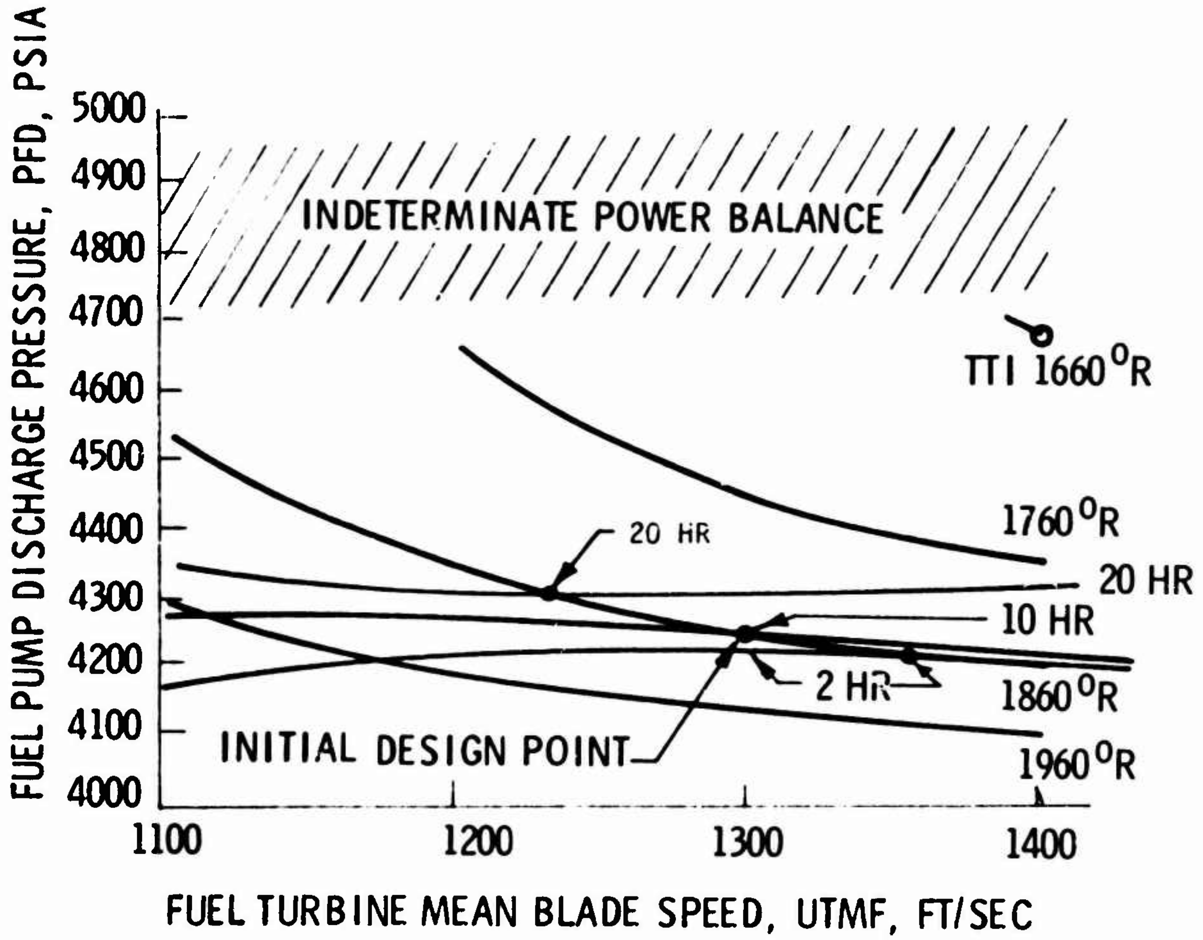
Fuel TPA bearing total thrust load capability is reduced from 75 pounds to 20 pounds with run time increased from 10 hours to 20 hours, as shown in Table LXVIII. Since a bearing will require 20 to 25 pounds preload, this bearing will not be capable of accommodating an external load. Therefore, a hydrostatic thrust bearing would be required. The load carrying capability can be increased to 65 pounds by reducing shaft speed to 70,000 rpm. The oxidizer bearing thrust load capability is reduced from 100 to 80 pounds with run time increased from 10 to 20 hours.

f. Gimbal Angle and Acceleration Rate

The effect of gimbal angle requirements on gimbal actuator power requirement and gimbal actuator weight was established and is discussed in the following section.

The modifications of gimbal angle requirements does not impact the engine concept and configuration.

Thrust vector control (TVC) actuator power and weight data have been generated for the 25,000 lb OOS engine parametric study.



- F = 25K
- P_c = 1800 PSIA
- MR = 6:1
- STAGED COMBUSTION
- UTMO = 1100 FT/SEC
- TTI = TTIF = TTIO
- BURN MIX = ONE 1000 SEC + FIVE 40 SEC BURNS
- REF. RUN 27 JULY 71 12:12:55.702

Figure 244. OOS Fuel Pump Required Discharge Pressure

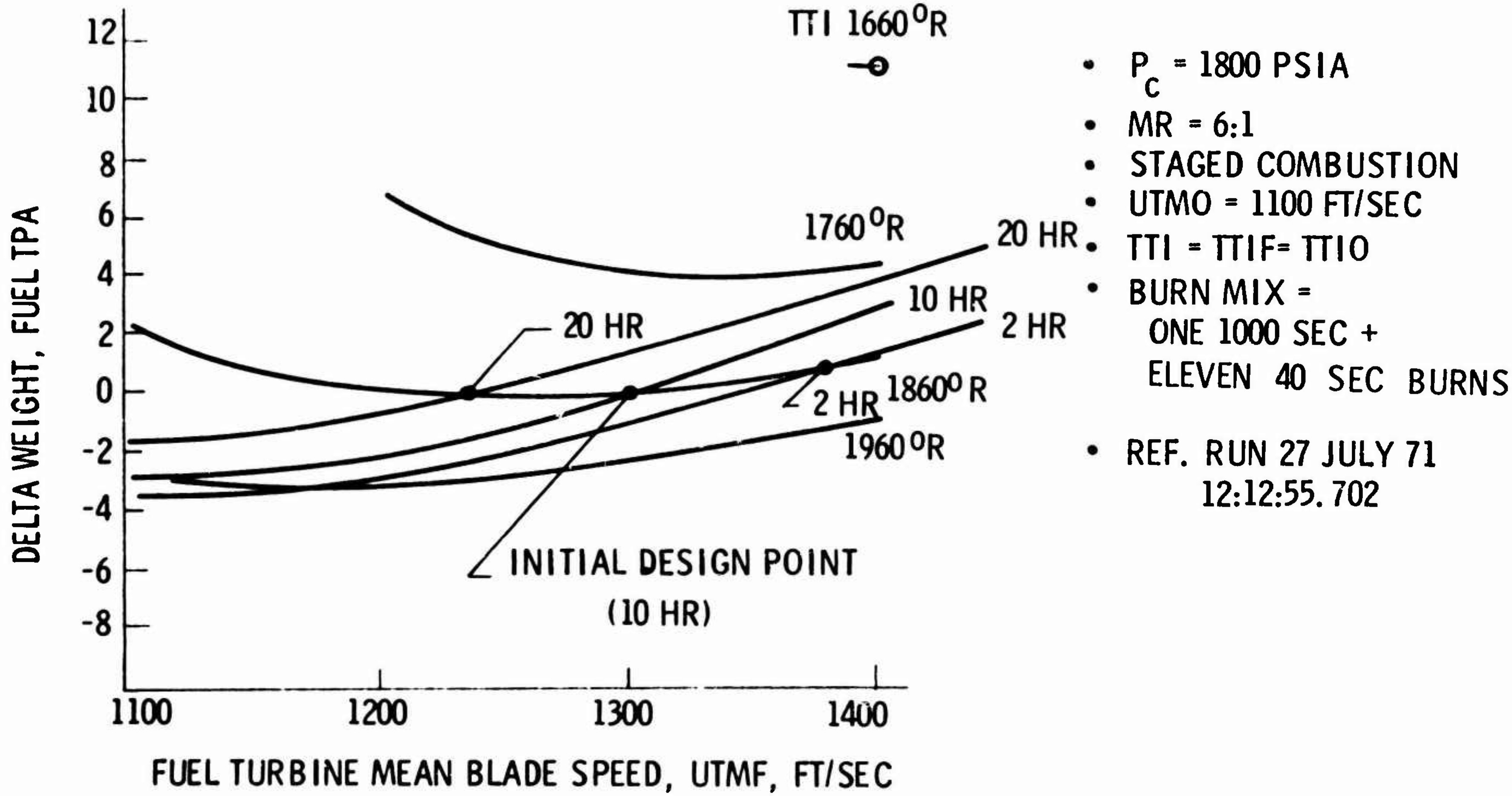


Figure 245. Fuel Turbopump Weight Sensitivity

TABLE LXIX

SUMMARY OF CYCLE LIFE CAPABILITY FOR REQUIRED DURATIONS

	Required Duration per Overhaul		
	2 Hours	10 hr (Baseline)	20 Hours
Main Combustion Chamber	1,220 cycles	1,220 cycles	1,220 cycles
Regen-Cooled Nozzle Tubes	10,200 cycles	10,200 cycles	10,200 cycles
Main Injector Vanes - OFHC CY	13,200 at 10 hr Factor of Safety = 8.80	9,800 at 50 hr Factor of Safety = 6.53	5,500 at 100 hr Factor of Safety = 3.67
Preburner U-Tube Liner	8,800 cycles	8,800 cycles	8,800 cycles
Nozzle Skirt Extension-AGcarb	Lose 4 mils in 10 hr(1)	Lose 21 mils in 50 hr(1)	Lose 42 mils in 100 hr(1)

(1) Max skirt regression rate = 0.425 mils/hr when uncooled. Cycle life capability is unaffected.

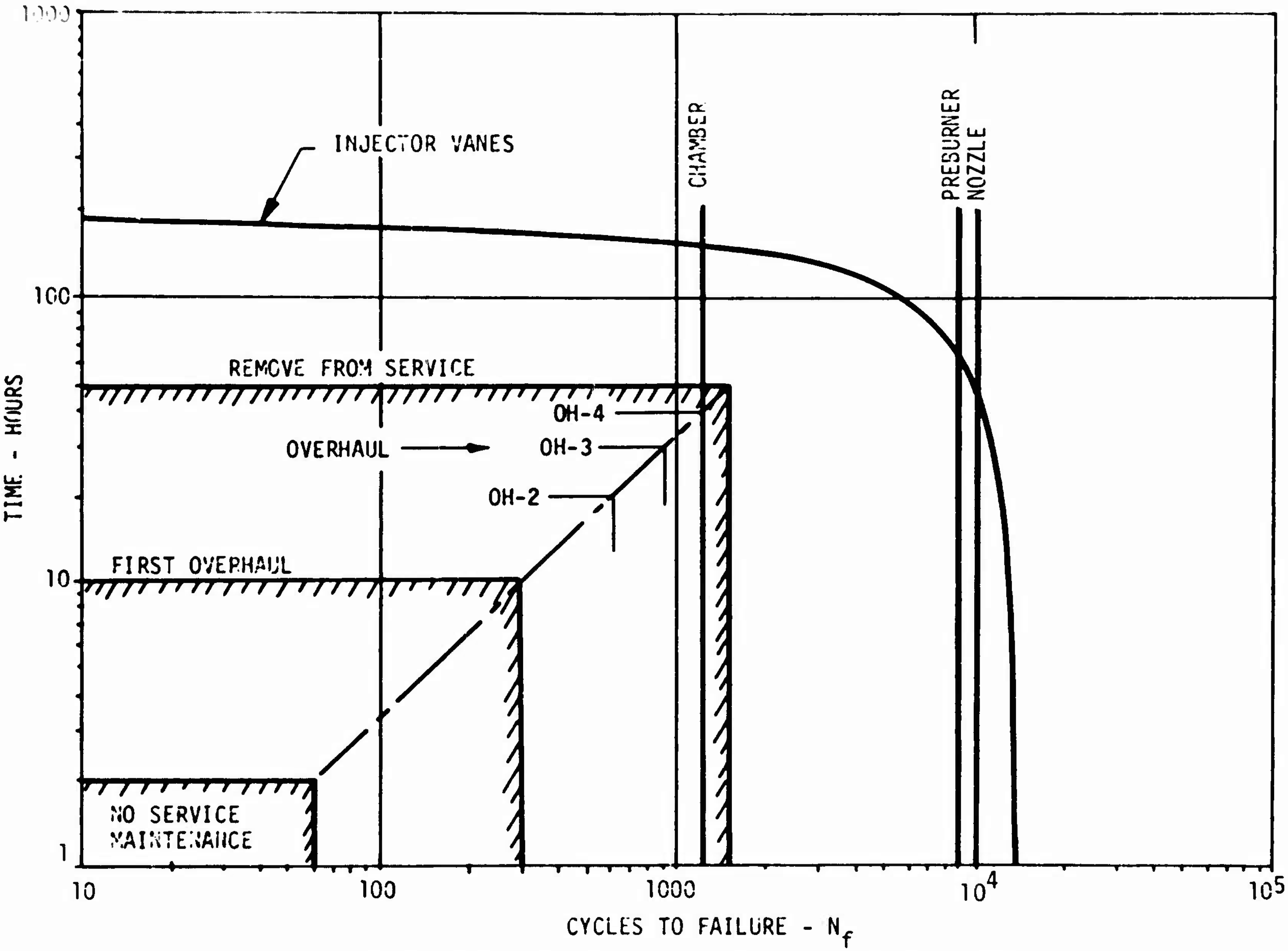


Figure 247. Combustion Components Service Life Capability

Variations in maximum gimbal angle, gimbal acceleration, and nozzle area ratio are considered for fixed and multi-position nozzles. Presented below are a summary of results, approach and assumptions used in the study, and a listing of pertinent numerical inputs.

A summary of parametric study results is given in Table LXX. The baseline engine designs for fixed and retractable nozzles utilize area ratios of 270 and 450, respectively. Required actuator power values are not large, and it is thus possible that electrical actuators would be more attractive than the hydraulic systems assumed for this study.

The approach to this study is, first, to assume that the TVC actuation system type is hydraulic. Next, the various contributions to the actuator "bias load" (total force which must be reacted by the actuator) are identified and combined. The actuator is then sized to handle the bias load and also checked against the requirement for a minimum TVC system natural frequency (in all cases bias load is found to be the dominant consideration). Finally, actuator weight, power, etc. are computed.

The following bias load contributions are considered:

(1) Thrust Misalignment (M_1)

A misalignment of the thrust vector (25,000 lb) of 0.25 inch is assumed, yielding a moment of 6250 in.-lb for all configurations.

(2) Gimbal Friction (M_2)

A sliding contact gimbal is assumed with 0.05 coefficient of friction and 5 inch radius. With 25,000 lb thrust, this gives 6250 in.-lb moment (same for all configurations).

(3) Duct Stiffness (M_3)

The stiffness of the articulating propellant lines between the low and high pressure pumps produces a moment proportional to gimbal displacement. Based on scaled results from Titan and Space Shuttle designs, the stiffness is estimated at K_θ = 400 in.-lb/degree. The bias moment contribution with the maximum gimbal angle (θ_{max}) is then

$$M_3 \text{ (in.-lb)} = K_\theta \times \theta_{max} \text{ (deg)}$$

(4) Lateral Vehicle Acceleration (M_4)

The lateral vehicle acceleration produces a moment about the gimbal point as follows:

TABLE LXX

SUMMARY OF TVC ACTUATOR PARAMETRIC STUDY

Fixed Nozzle

Gimbal Acceleration (rad/sec^2)	5 →														10
Nozzle Area Ratio	180 →			270 →			400 →			450 →			500 →		270
Maximum Gimbal Angle (deg)	3	5	10	3	5	10	3	5	10	3	5	10	3	5	10	5
Actuator Delivered Power[(1)] (HP)	0.152	0.155	0.162	0.164	0.157	0.178	0.186	0.190	0.203	0.204	0.207	0.219	0.216	0.219	0.233	0.183
Actuator Weight[(2)] (lbm)	16.5	17.2	19.1	16.6	17.3	19.4	16.7	17.6	20.0	16.8	17.8	20.3	16.9	17.9	20.6	17.9
Stall Force (lb)	1620	1660	1730	1760	1790	1900	1990	2040	2180	2190	2220	2350	2320	2350	2490	1960
Stroke (in.)	0.964	1.604	3.208	0.964	1.604	3.208	0.964	1.604	3.208	0.964	1.604	3.208	0.964	1.604	3.208	1.604
Cylinder Diameter (in.)	1.410	1.420	1.440	1.445	1.455	1.486	1.510	1.520	1.550	1.555	1.563	1.592	1.587	1.592	1.623	1.500

"Minimum Weight," Retractable Nozzle

Gimbal Acceleration (rad/sec^2)	5 →									10
Nozzle Area Ratio	400 →			450 →			500 →			450
Maximum Gimbal Angle (deg)	3	5	10	3	5	10	3	5	10	5
Actuator Delivered Power[(1)] (HP)	0.291	0.295	0.315	0.318	0.327	0.343	0.333	0.338	0.357	0.402
Actuator Weight[(2)] (lbm)	16.9	18.7	22.4	17.6	19.0	23.0	17.7	19.2	23.3	19.9
Stall Force (lb)	3114	3153	3377	3400	3455	3667	3561	3615	3827	4300
Stroke (in.)	0.964	1.604	3.208	0.964	1.604	3.208	0.964	1.604	3.208	1.604
Cylinder Diameter (in.)	1.76	1.77	1.81	1.82	1.83	1.87	1.85	1.86	1.90	1.98

(1) Maximum delivered power for one actuator
(2) One Actuator

$$M_4 \text{ (in.-lb)} = W \times G_{lat} \times \bar{X}_{cg}$$

where W = engine gimbaled weight (lbm)

$\bar{X}_{cg}$ = longitudinal distance, gimbal point to c.g. of gimbaled components (in.)

G_{lat} = lateral vehicle acceleration = 1.0g

Engine mass properties are computed from component data. Vehicle accelerations are taken from Space Shuttle main engine specifications per coordination with customer (G_{lat} = 1.0, G_{long} = 3.0).

(5) Longitudinal Vehicle Acceleration (M_5)

The lateral c.g. offset and moment arm produced by gimbaling of the aft c.g. engine combine to produce a bias moment due to longitudinal vehicle acceleration.

$$M_5 \text{ (in.-lb)} = W \times G_{long} \left[\bar{Y}_{cg} + \bar{x}_{cg}\, \theta_{max} \text{ (rad)} \right]$$

where G_{long} = longitudinal vehicle accel. = 3.0 g's

$\bar{Y}_{cg}$ = lateral distance, gimbal point to c.g. of gimbaled components (in.)

(6) Gimbal Acceleration (M_6)

A maximum gimbal acceleration $\ddot{\theta}_{max}$ (rad/sec^2) acting on an engine inertia I (lb-sec^2-in.) yields a bias moment of

$$M_6 \text{ (in.-lb)} = I \times \ddot{\theta}_{max}$$

The design bias moment is developed by forming the root-sum-square of moments M_1, M_3, M_4, M_5, and M_6 (algebraic addition is considered overly conservative when considering maximum values for each contribution) and adding the gimbal friction moment M_2 (which is always present). The bias load is the bias moment divided by the moment arm.

The actuator is then sized to a stall force 1.5 times the bias value, the accepted practice in design of hydraulic actuators. Thus,

$$A_p = \frac{3}{2} \frac{M_{bias}}{P_s \ \ell}$$

where A_p = net piston area (in.^{2})

M_{bias} = bias moment (in.-lb)

ℓ = actuator moment arm = 9.2 in.

P_s = supply pressure = 3000 lb/in.^{2}

The sizing based on bias moment is then checked against the piston area required to obtain a 10 Hz "hydraulic" natural frequency (it is assumed that 10 Hz minimum hydraulic and structural frequencies combine to yield a 5 Hz minimum TVC system natural frequency). The minimum piston area is expressed as

$$(A_p)_{min} = \frac{(2\pi \times 10)^2 \ (1.05) \ \theta_{max} \ (\text{rad}) \times I}{2 \ \beta_E \ (\text{lb/in.}^2) \times \ell}$$

where β_E = effective bulk modules = 100,000 lb/in.^{2}

Maximum delivered actuator power is developed while operating at 2/3 supply pressure with maximum velocity. For 10 deg/sec maximum no-load gimbal rate, the power is expressed as

$$P_{max\ del} \ (\text{HP}) = 0.0305 \ A_p \ (\text{in.}^2) \ \ell \ (\text{in.})$$

Actuator weight estimates are made by scaling actual hardware on the basis of swept volume,

$$V_{swept} \ (\text{in.}^3) = X \times A_p$$

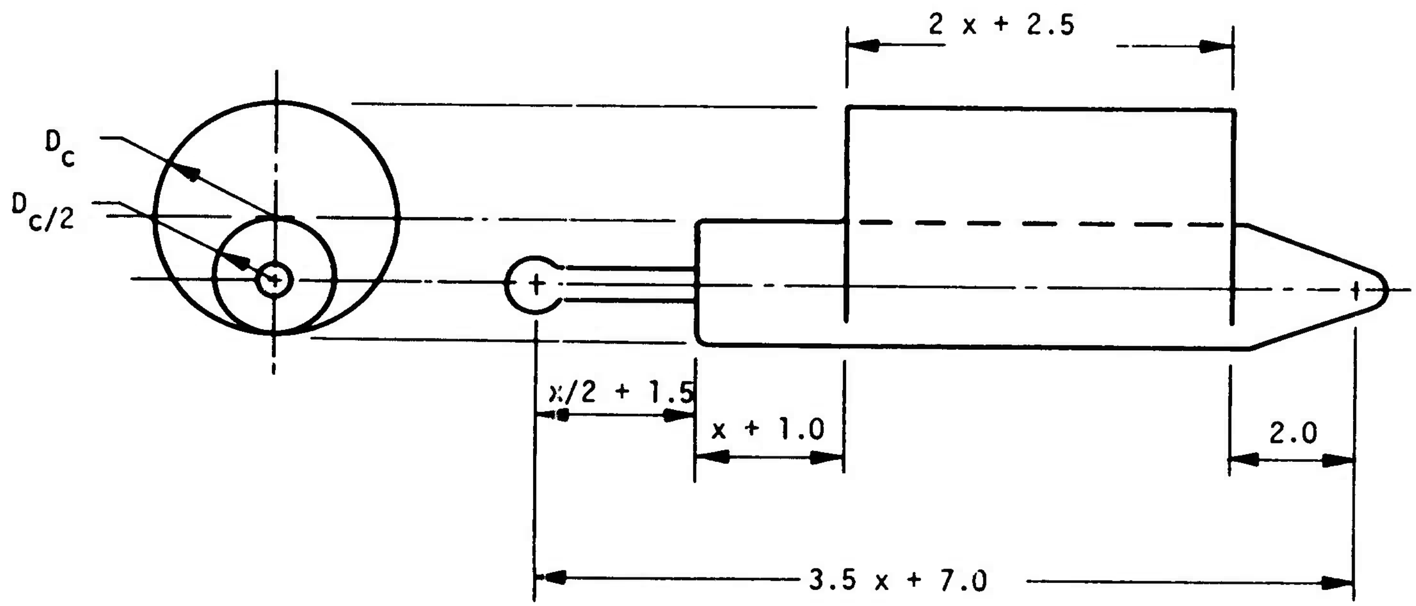
where X (in.) = stroke = $2 \times \theta_{max}$ (rad) $\times \ \ell$

The expression

$$W_{act} \ (\text{lbm}) = 15.5 + 1.92 \ V_{swept}$$

is derived from Titan III TVC actuator design data. These units employ redundant servo-valves to obtain fail-operate-fail-safe performance.

Actuator dimensions are estimated from existing designs. An envelope sketch is given in Figure 247 along with equations for cylinder OD and minimum length.



MINIMUM LENGTH = 3.5 x + 7.0 (IN.)

CYLINDER O.D. = D_c = 0.5 + 1.237 $\sqrt{A_p}$ (IN)

Figure 247. OOS Actuator Dimensions

Numerical values used in the parametric study are listed in Table LXXI. Sources of information are noted where applicable.

Figures 248 through 250 are plots of gimbal actuator power requirements with respect to nozzle exit area ratio, gimbal angle, and acceleration, Figure 251 depicts weight change with respect to gimbal angle.

g. Nozzle Area Ratio, Fixed and Retractable

The change of nozzle area ratios affect engine performance, weight, envelope and gimbal power requirements. The effect on engine weight due to area ratio is presented in Figure 252 for MR = 6.0 for all regen cooled engine concepts and graphite radiation cooled engine concept for fixed and retractable nozzles. This figure indicates that graphite nozzle extensions can result in considerable weight saving, if large area ratio nozzles are used. In the presented data it was assumed that the transition area ratio from regen cooled to radiation cooled concept is at ε = 125:1. For small area ratio nozzles the weight savings with radiation cooled nozzles are insignificant.

The performance impact due to area ratio is shown in Figure 253 for constant chamber pressure of 1800 psia and ERE = 99% at MR = 6.0 and relates specific impulse gains vs specific impulse. The engine envelope impact is shown in Figure 254 presenting engine overall length and maximum engine diameter.

For all regen cooled chambers, the nozzle expansion area ratio has an impact on chamber cooling and thermal chamber life. The larger the area ratio, the larger the chamber pressure drop and coolant temperature result in increased thrust chamber gas-side wall temperature reducing the chamber life capability. The impact of the nozzle area ratios on coolant conditions is shown in Figure 255.

Radiation cooled nozzles have the advantage that the impact of area ratio on coolant conditions and life is very small since the impact is only due to contour changes assuming a fixed transition area ratio.

The retractable nozzle is radiation cooled fibrous graphite material. The retraction mechanism consists of cable pully and drive. A comparison of fixed and retractable nozzles is shown in Figures 256 and 257 for the two missions considered.

In these figures, the payload capability of the fixed and retractable nozzles is compared indicating very small payload gains of the retractable nozzle for both missions. For the orbit-to-orbit mission, the optimum retractable area ratio is smaller than for the lunar lander mission due to the lesser sensitivity of weight.

TABLE LXXI

NUMERICAL VALUES USED IN GIMBAL ACTUATOR PARAMETRIC STUDY

Thrust = 25,000 lb

Max Gimbal Angle (nominal) = 5 degrees

Max Gimbal Rate = 10 deg/sec[1]

Max Gimbal Acceleration (nominal) = 5 rad/sec^2

Structural Natural Frequency, minimum = 10 Hz[1],[2]

TVC System Bandwidth, minimum = 5 Hz[1]

Longitudinal Vehicle Acceleration = 3.0 g's[1]

Lateral Vehicle Acceleration = 1.0 g[1]

Inlet Line Stiffness = 400 in.-lb/deg[3]

Gimbal Block Radius = 5.0 in.

Gimbal Block Coefficient of Friction = 0.05[3]

Actuator Moment Arm = 9.2 in.

Thrust Misalignment = 0.25 in.[3]

Hydraulic Supply Pressure = 3000 lb/in.^{2}[3]

Effective Bulk Modulus = 100,000 lb/in.^{2}[3]

Mass Properties (wet, gimbaled components)

Nozzle Type	Fixed					Retr', Min Weight		
Area Ratio	180	280	400	450	500	400	450	500
Weight (lbm)	315.3	332.6	354.4	365.7	374.5	403.8	415.1	423.9
$\overline{X}_{cg}$ (in.)	-15.2	-18.2	-22.5	-25.0	-26.6	-22.1	-24.3	-24.9
$\overline{Y}_{cg}$ (in.)	1.8	1.7	1.6	1.6	1.5	1.2	1.2	1.1
I (lb-sec^2-in.)	445	697	1180	1495	1730	1208	1564	1800

(1) Negotiated with customer
(2) Natural frequency with actuator stiffness = ∞
(3) Assumed on basis of past experience

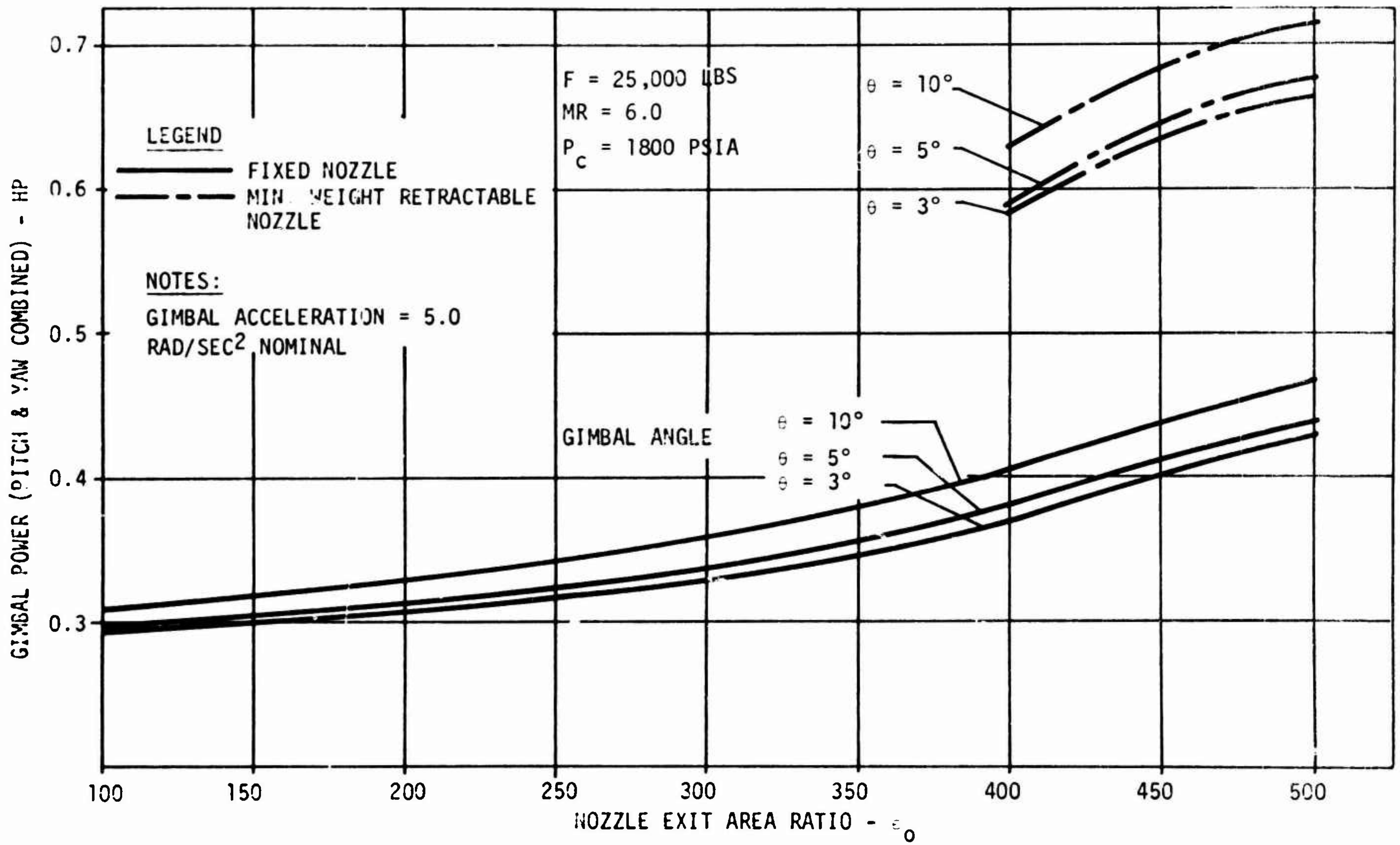


Figure 248. Gimbal Power vs Nozzle Exit Area Ratio

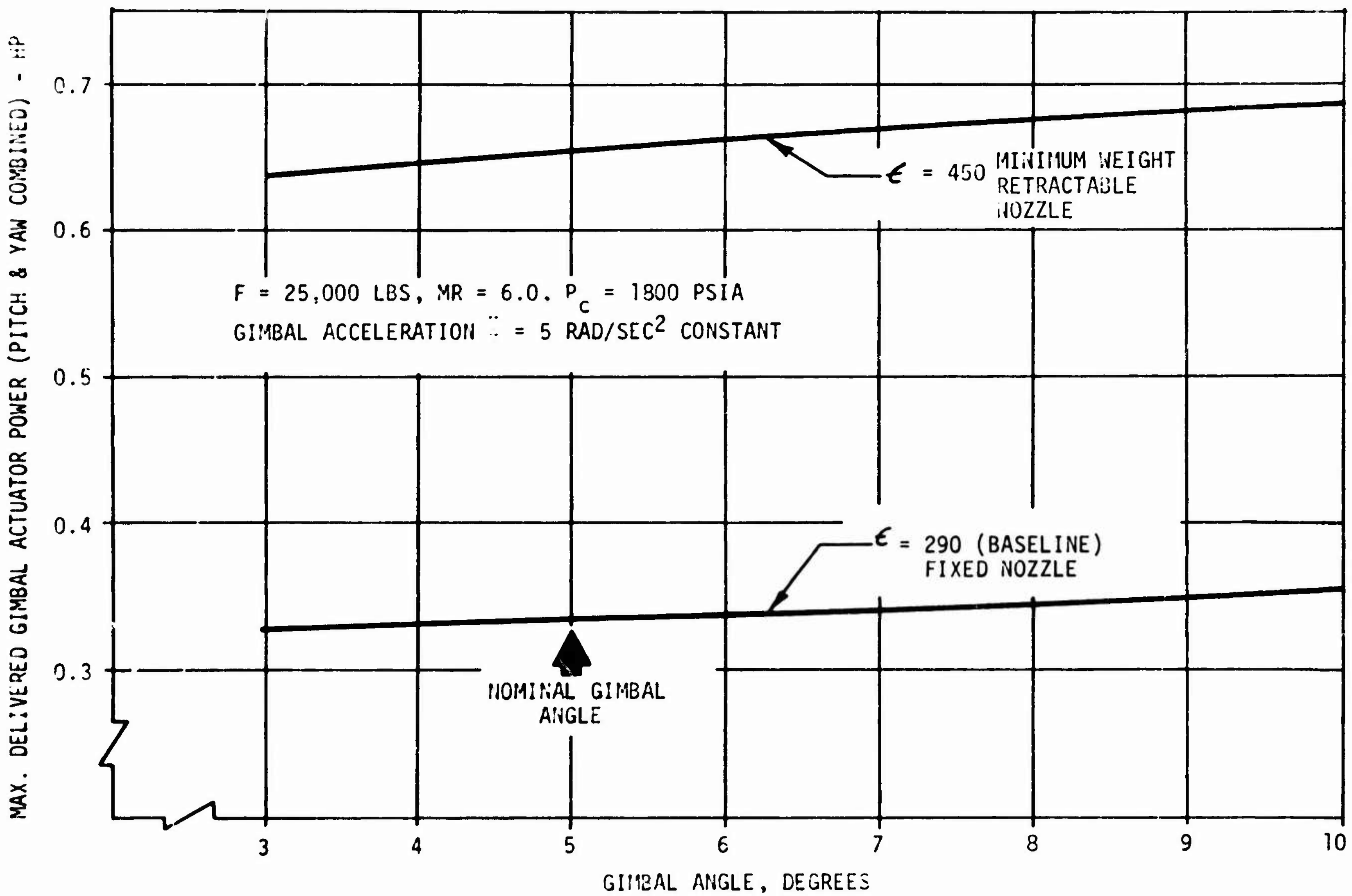


Figure 249. Gimbal Power vs Gimbal Angle

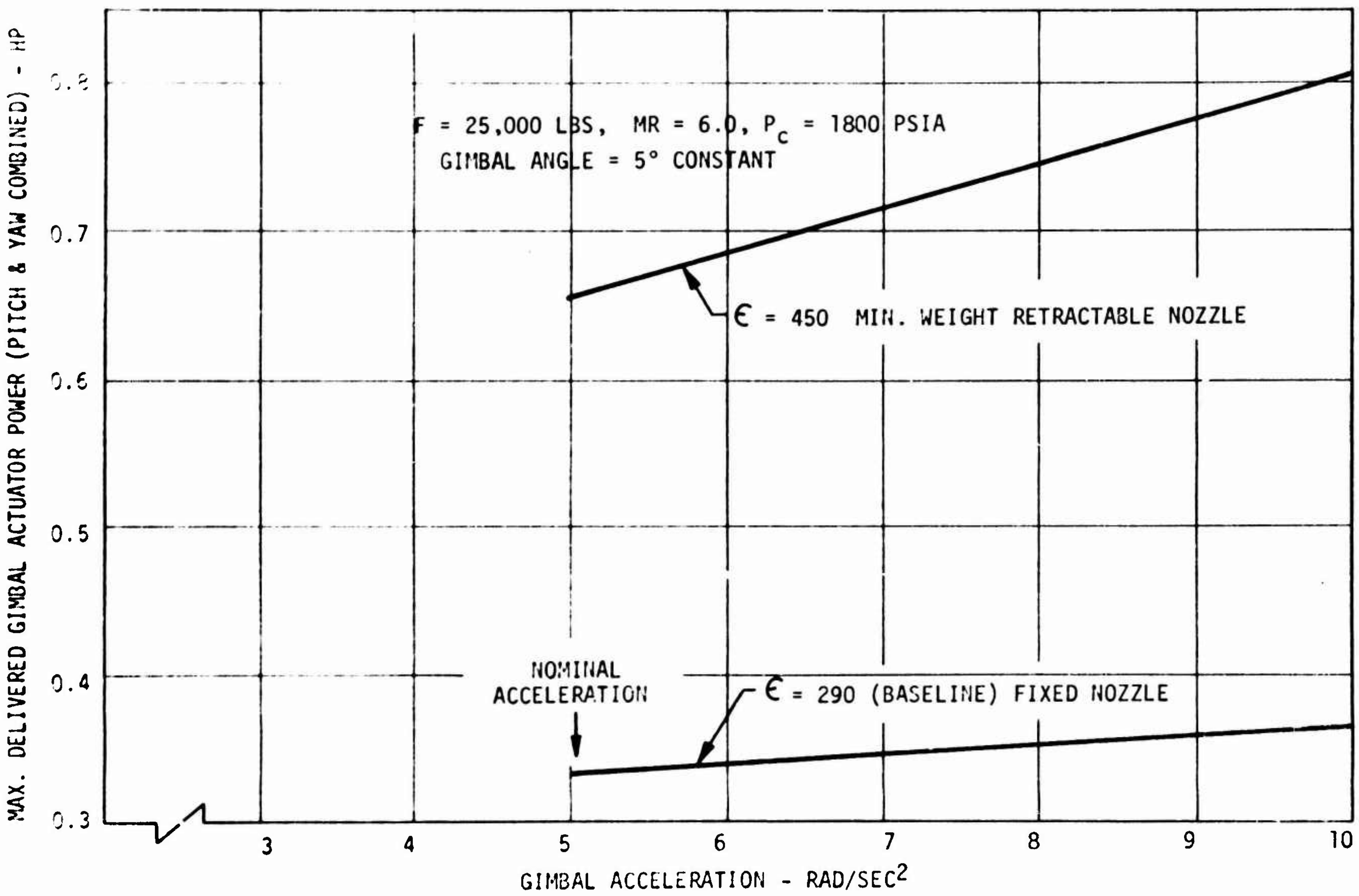


Figure 250. Gimbal Power vs Gimbal Acceleration

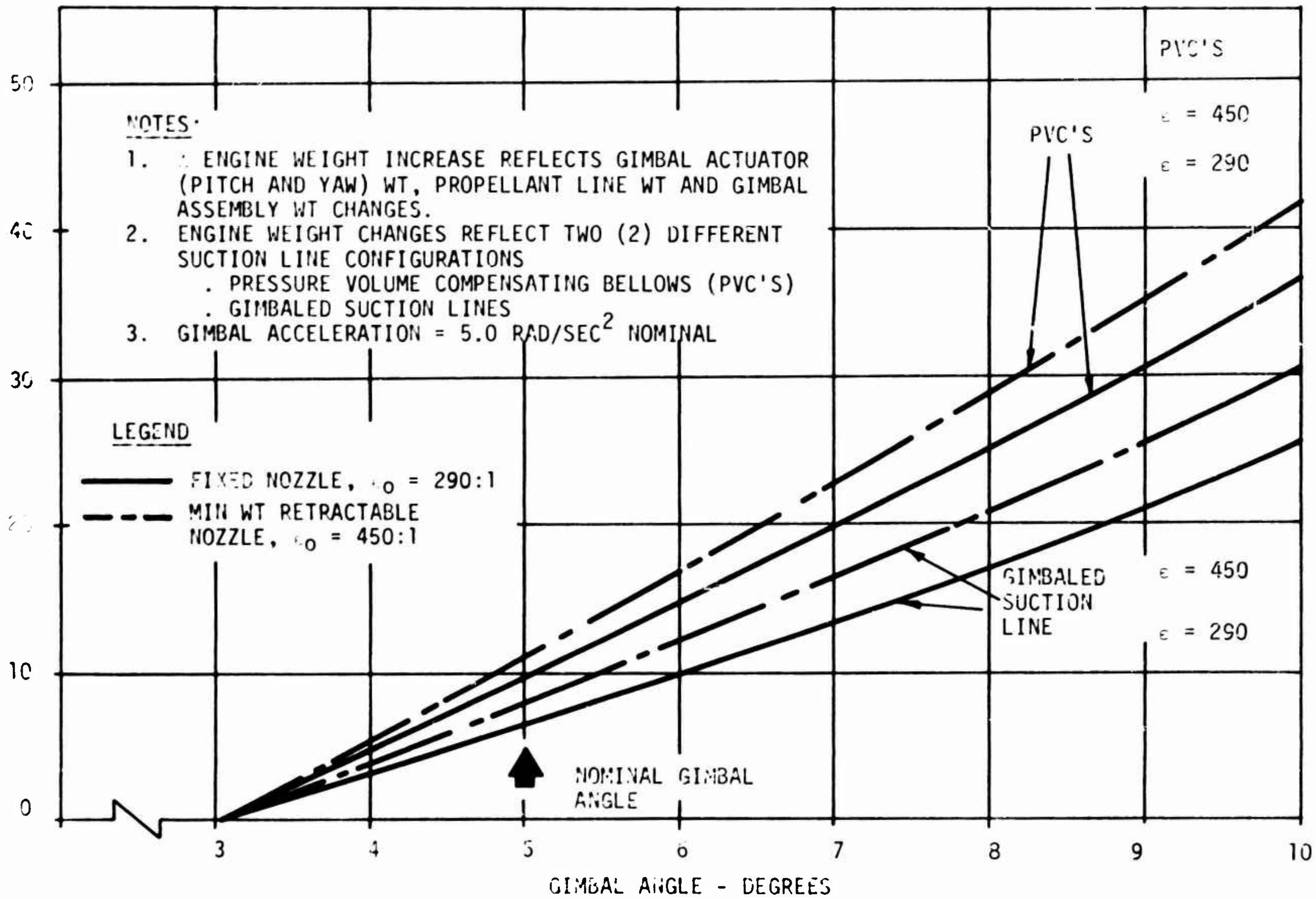


Figure 251. Engine Weight Increase vs Gimbal Angle

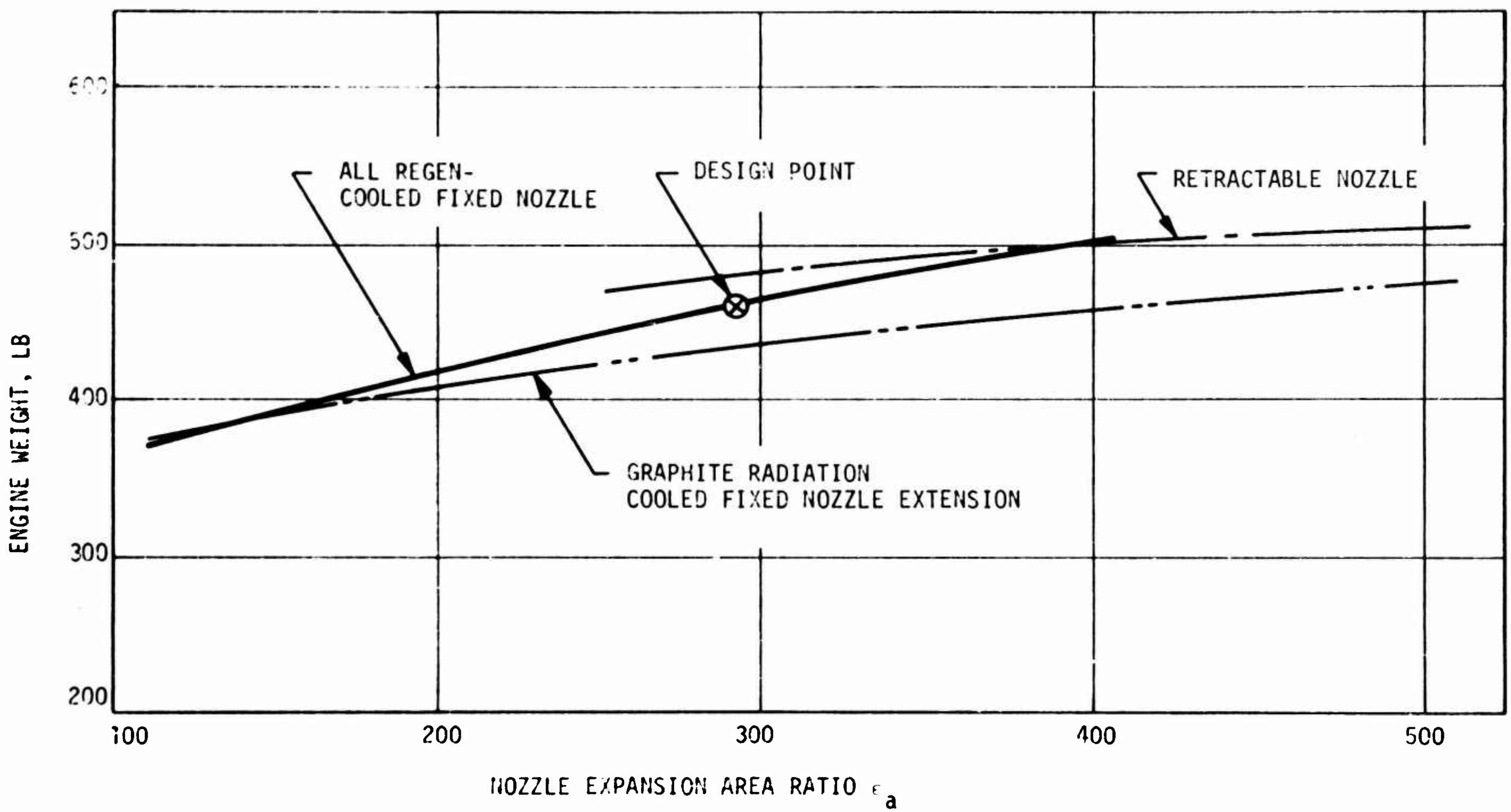


Figure 252. Engine Weight vs Nozzle Exit Area Ratio

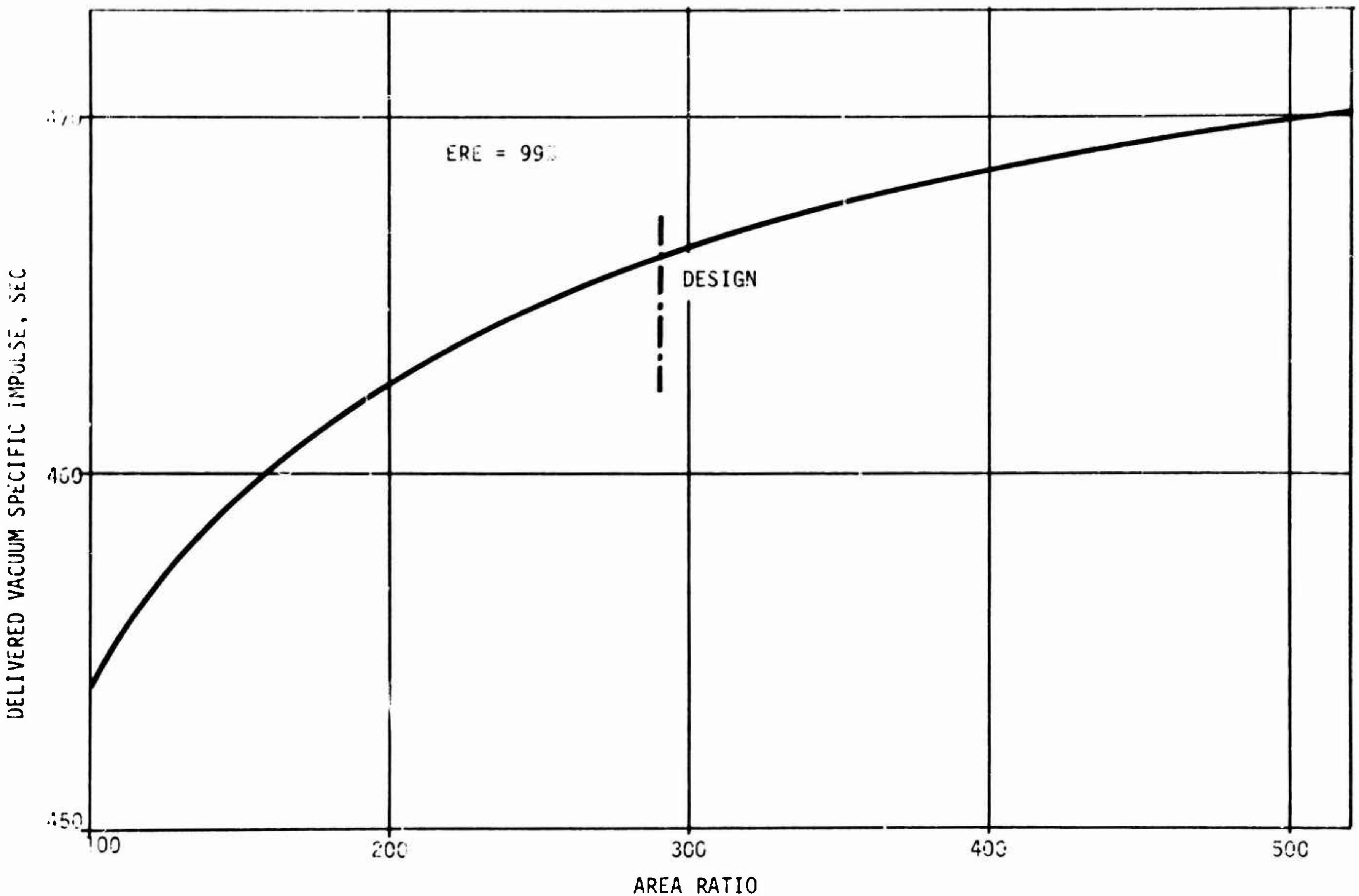


Figure 253. Performance vs Nozzle Exit Area Ratio

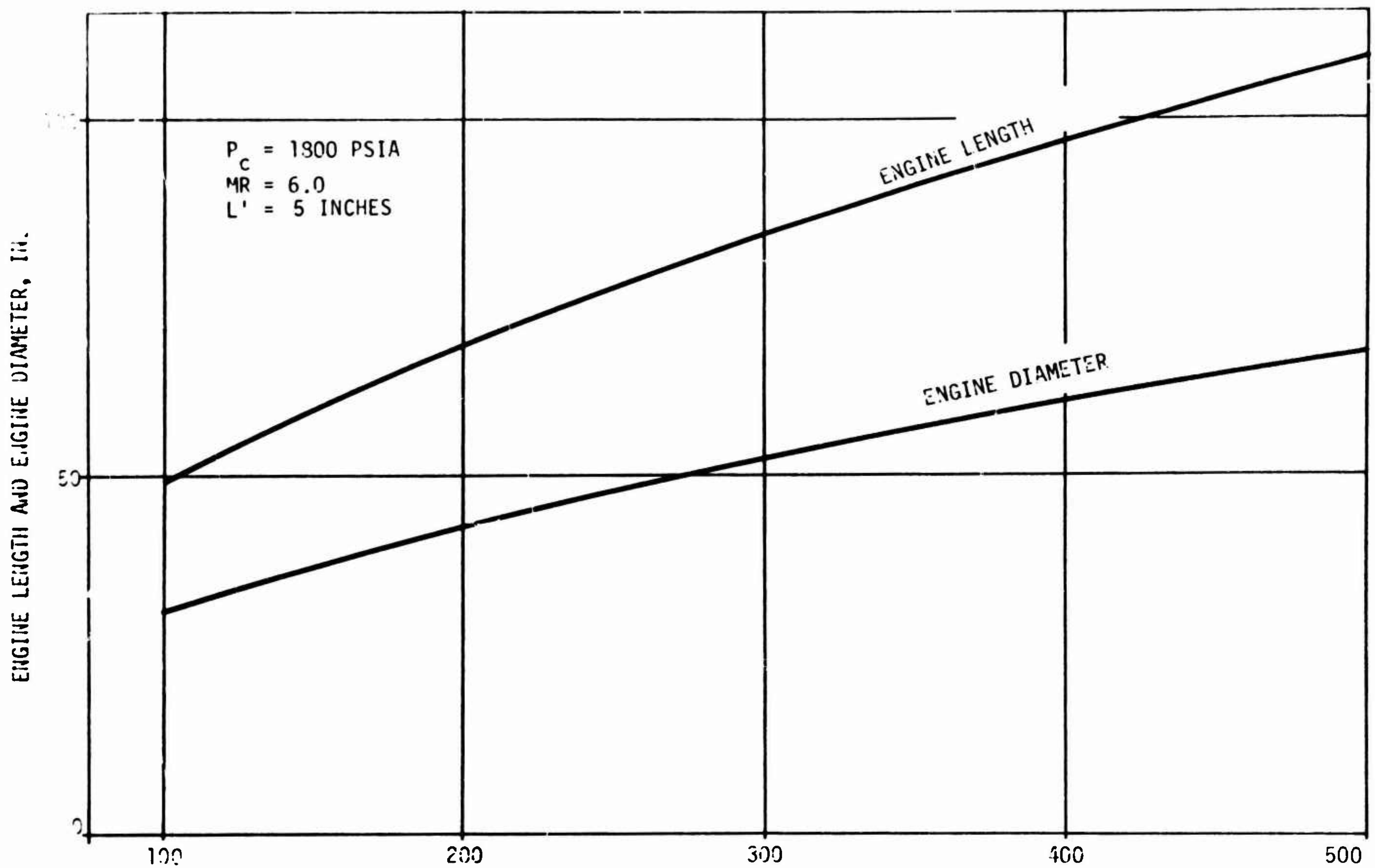


Figure 254. Engine Envelope vs Nozzle Exit Area Ratio

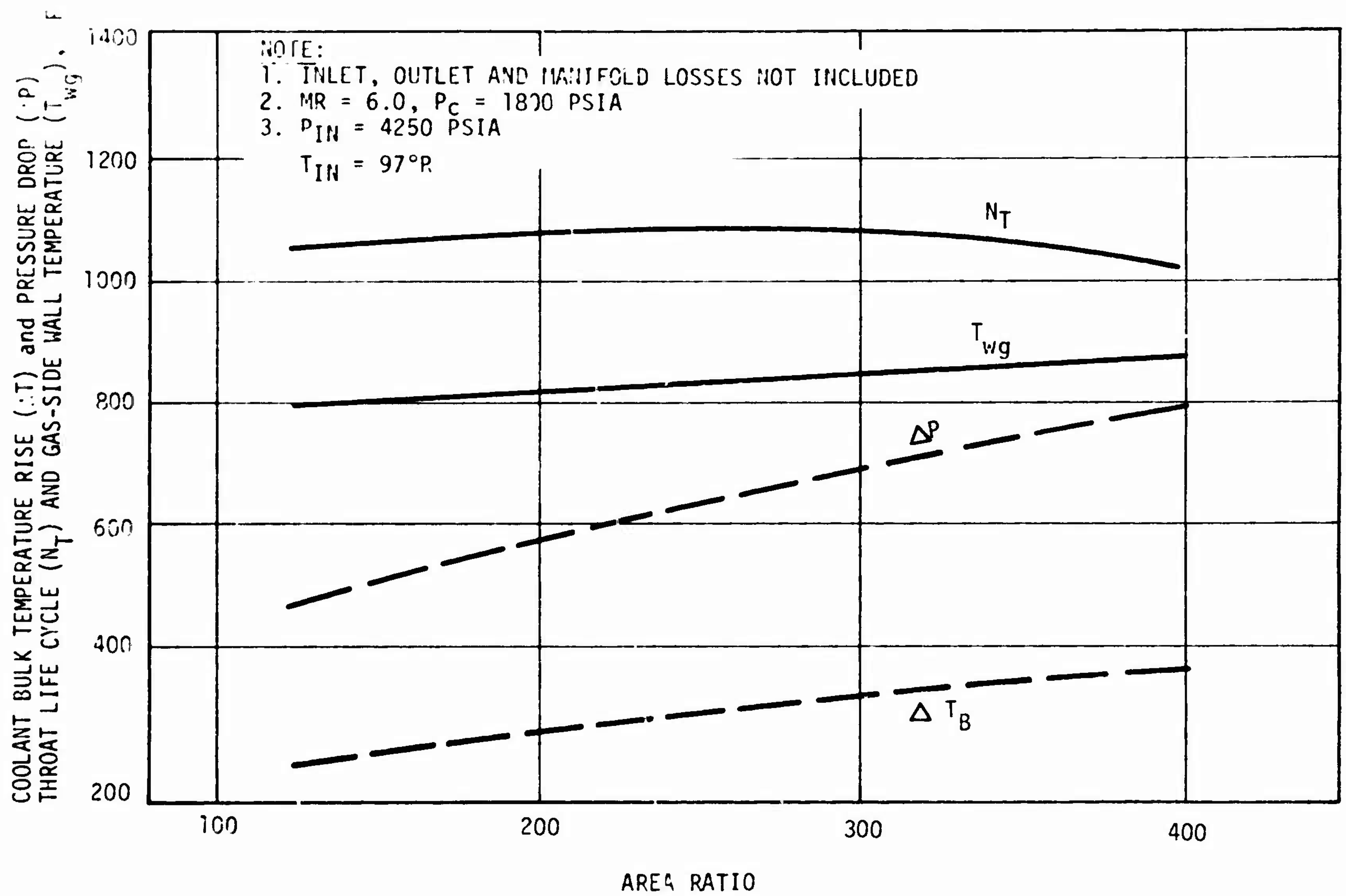


Figure 255. Nozzle Area Ratio vs Coolant Conditions

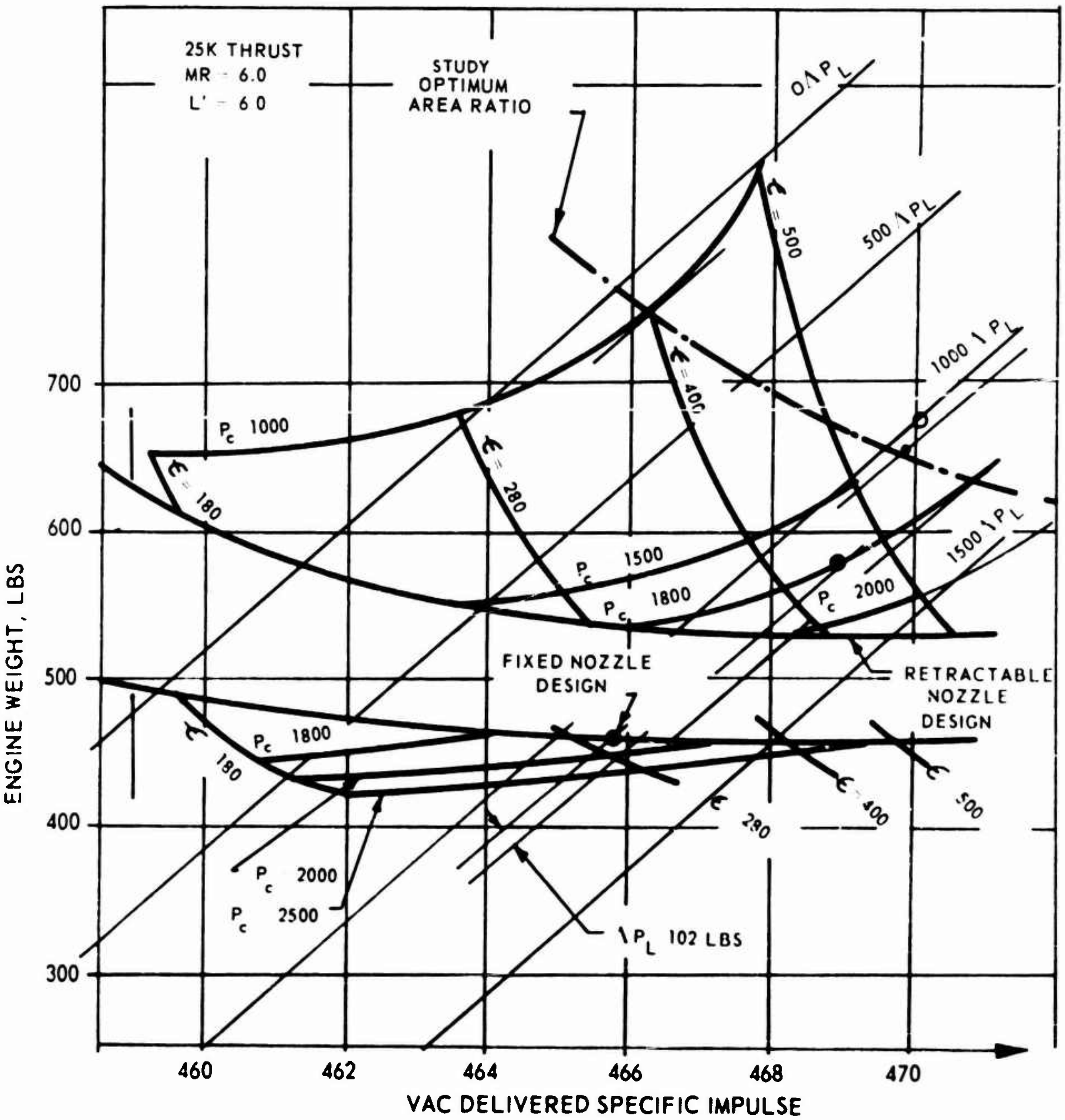


Figure 256. Fixed and Retractable Nozzle Comparison Orbit-to-Orbit Mission

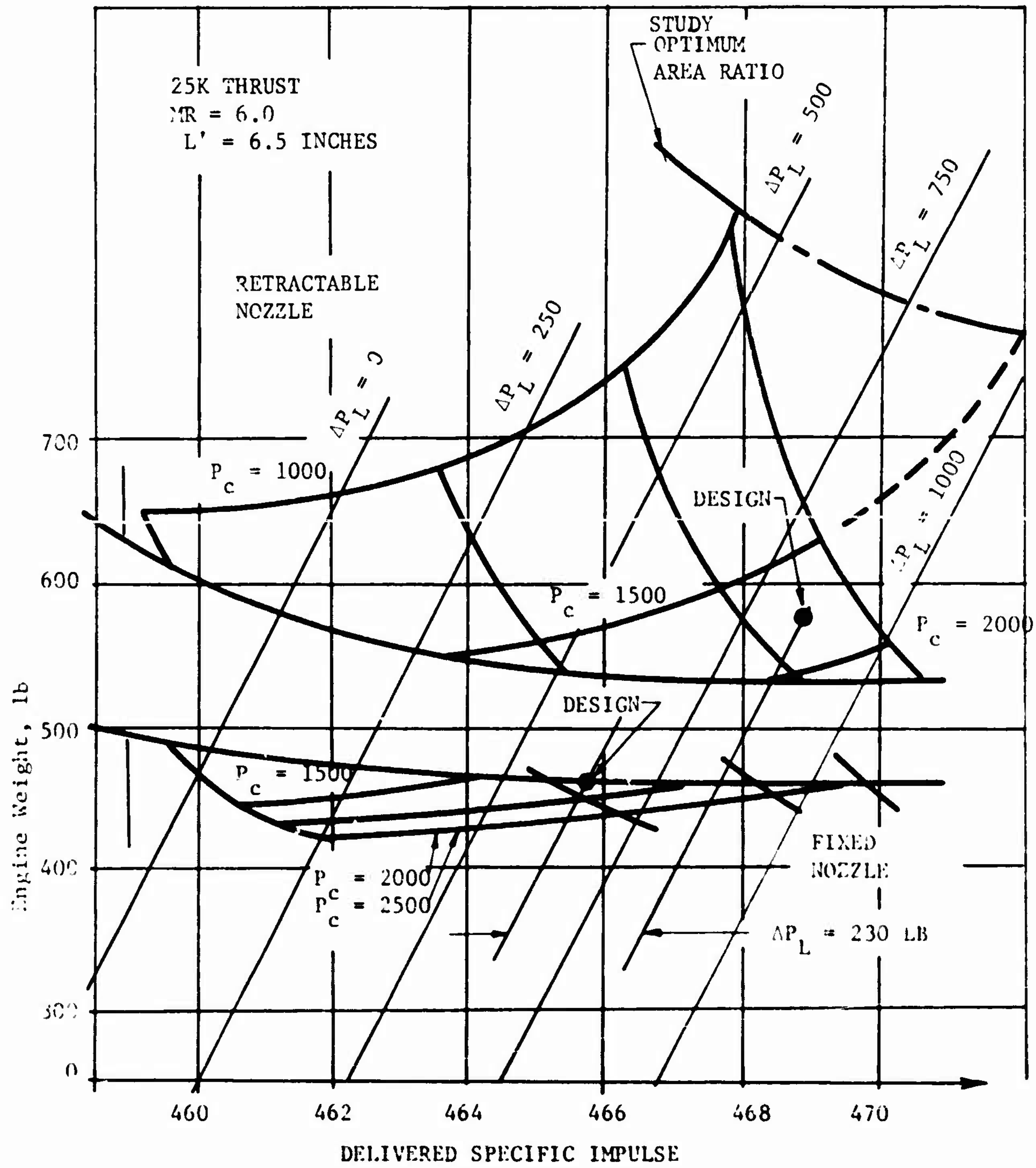


Figure 257. Fixed and Retractable Nozzle Comparison, Lunar Lander Mission

The retractable nozzles are not designed at optimum mixture ratio due to the relatively small gains in payload, and as a design area ratio for retractable nozzles of 450:1 is recommended. Since the payload gains are very small for retractable nozzles, only fixed nozzles are considered for the 82-in. envelope. The small loss on payload is considered a good trade r reduced systems complexity, maintenance, and reliability considerations.

h. Suction Conditions

The matrix of suction condition variations for mixture ratio MR = 6.0 is shown in Table LXXII and Table LXXIII.

The effect of NPSH on engine design conditions were previously evaluated during the parametric engine design analysis. The suction requirement only effected engine weight to a very small degree since only the boost pumps are affected. Engine performance is not affected by changing NPSH requirements. Engine envelope and data are also insensitive. The suction line diameters are, however, changing with suction requirements.

The boost pumps were designed to operate with the specified 16 and 60 ft oxidizer and fuel minimum NPSH values. Boost pumps were evaluated with the main TPA for operation with reduced values of NPSH down to zero feet. The results of varying fuel NPSH are given in Table LXXII with cases 1 through 5 showing the effect of designing for fuel NPSH values of 0, 15, 30, and 60 feet while holding the oxidizer NPSH constant at 16 feet. The results of varying oxidizer NPSH are given in Table LXXIII with cases 6 through 10 showing the affect of designing for oxidizer NPSH values of 0, 2, 4, 8 and 16 feet while holding the fuel NPSH constant at 60 feet. The impact on the system power balance pressure level, envelope, and weight are shown.

The affect of designing the fuel propellant feed system to accommodate NPSH values down to zero feet are insignificant. Designing the oxidizer propellant feed system to accommodate NPSH values of zero feet does increase the system power balance pressure by 44 psi, increases the oxidizer and fuel TPA weight by 4.8 and 0.9 lb, respectively, and increase also the oxidizer TPA length by 7-6-in. The effect of NPSH on both fuel and oxidizer suction lines is shown in Figure 258.

i. Run Duration

The engine sensitivity to single burn duration is limited to the turbine disk and turbopump bearings and does not effect other components.

TABLE LXXII

OOS 25K ENGINE
FUEL PUMP NPSH SENSITIVITY

	CASE 1		CASE 2		CASE 3		CASE 4		CASE 5*	
	OXID	FUEL	OXID	FUEL	OXID	FUEL	OXID	FUEL	OXID	FUEL
NPSH, FEET	16	0	16	15	16	30	16	45	16	60
PUMP DISCHARGE PRESSURE	3092**	4257	3092**	4255	3092**	4255	3092**	4255	3092**	4255
BOOST PUMP SHAFT SPEED	14286	22857	14286	22857	14286	22857	14286	22857	14286	22857
SUCTION SPECIFIC SPEED	25742	16781	25742	15455	25742	14352	25742	13412	25742	12610
SUCTION LINE DIAMETER	3.135	3.604	3.135	3.604	3.135	3.604	3.135	3.604	3.135	3.604
DELTA LENGTH, TPA + BOOST	0.0Δ	+0.002	0.0Δ	+0.001Δ	0.0Δ	+0.001Δ	0.0Δ	0.0Δ	Ref.	Ref.
DELTA WEIGHT, BOOST TPA	0.0Δ	-0.020Δ	0.0Δ	-0.015Δ	0.0Δ	-0.010Δ	0.0Δ	-0.005Δ	Ref.	Ref.
DELTA WEIGHT, MAIN TPA	+0.007Δ	+0.073Δ	+0.003Δ	+0.039Δ	+0.002Δ	+0.026Δ	+0.001Δ	+0.013Δ	Ref.	Ref.
DELTA WEIGHT, BOOST + MAIN	+0.007Δ	+0.053Δ	+0.003Δ	+0.024Δ	+0.002Δ	+0.016Δ	+0.001Δ	+0.008	Ref.	Ref.

* TASK I BASELINE

** FIRST STAGE PUMP

TABLE LXXIII

OOS 25K ENGINE
OXIDIZER PUMP NPSH SENSITIVITY

	CASE 6		CASE 7		CASE 8		CASE 9		CASE 10	
	OXID	FUEL	OXID	FUEL	OXID	FUEL	OXID	FUEL	OXID	FUEL
NPSH, FEET	0	60	2	60	4	60	8	60	16	60
PUMP DISCHARGE PRESSURE	3092**	4299	3092**	4267	3092**	4256	3092**	4249	3092**	4255
BOOST PUMP SHAFT SPEED	4671	22857	6331	22857	7856	22857	10650	22857	14286	22857
SUCTION SPECIFIC SPEED	28140	12610	28140	12610	28140	12610	28140	12610	25742	12610
SUCTION LINE DIAMETER	4.550	3.604	4.112	3.604	3.826	3.604	3.457	3.604	3.135	3.604
DELTA LENGTH TPA + BOOST	+7.645Δ	+0.023Δ	+4.760Δ	+0.006Δ	+3.142Δ	+0.001Δ	+1.336Δ	-0.003Δ	Ref.	Ref.
DELTA WEIGHT, BOOST TPA	+4.584Δ	0.0Δ	+2.708Δ	0.0Δ	+1.683Δ	0.0Δ	+0.610Δ	0.0Δ	Ref.	Ref.
DELTA WEIGHT, MAIN TPA	+0.198Δ	+0.896Δ	+0.114Δ	+0.251Δ	+0.016Δ	+0.035	-0.057Δ	-0.122Δ	Ref.	Ref.
DELTA WEIGHT, BOOST + MAIN	+4.782	+0.896Δ	+2.822Δ	+0.251Δ	+1.699Δ	+0.035Δ	+0.553Δ	-0.122Δ	Ref.	Ref.

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* INITIAL DESIGN

** FIRST STAGE PUMP

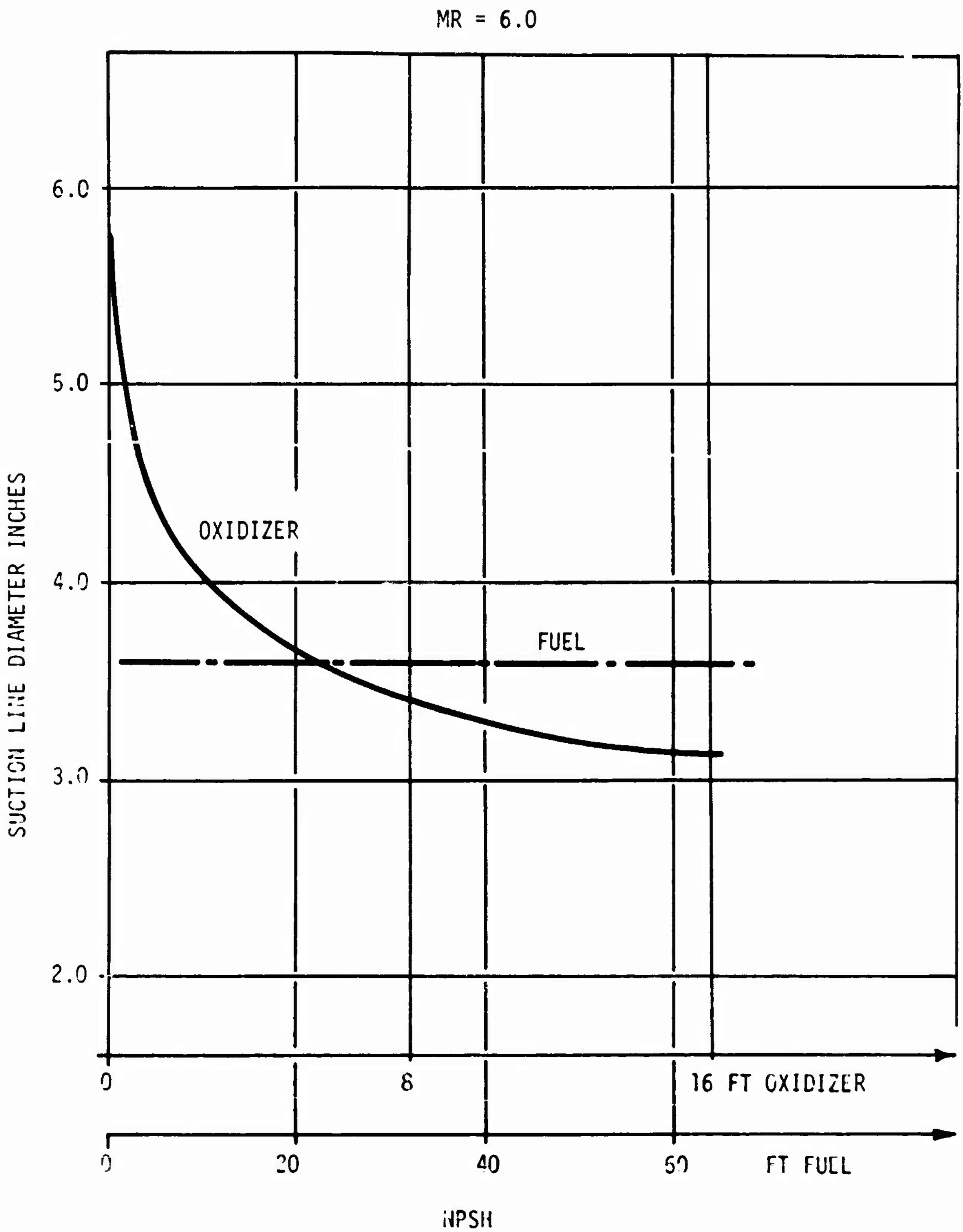


Figure 258. Suction Line Diameter vs NPSH

(1) Turbine Disks

Turbine operational life based on creep damage is based on the operational time that the turbine disk is at temperature. For a rim and side-heated disk, its temperature approaches gas temperature after about 100 seconds. For the baseline design with a 1000 sec duration long burn, the disk will be at temperature for 900 seconds. The 40 sec short burn is sufficiently short that the disk does not reach the gas temperature and therefore does not experience the high temperature creep damage. Thus, the disk is at temperature for 62.5% or 6.25 hours of the 10 hours design life. Varying the design long burn duration from 500 to 2000 sec while holding the cycle life at 300 with one long plus eleven 40 sec short burns varies the total time that the disk is at temperature. The disk exposed to a 500 sec duration long burn would accumulate 2.78 hours at temperature out of 6.53 hours operation whereas the disk exposed to 2000 sec duration long burns would accumulate 13.4 hours at temperature out of 16.94 hours. The sensitivity to long burn durations of 500, 1000 and 2000 seconds are shown in Figures 259, 260 and 261 which show fuel pump required discharge pressure, fuel turbopump and oxidizer turbopump weight sensitivities. The effects of run duration are also given in Table LXXIV which shows the effect on TPA interface dimensions and bearing maximum thrust loads capability. It should be noted that operation with a turbine gas temperature above 1860°R will result in a cycle life below the required 300.

(2) Turbopump Bearings

Fuel bearing total thrust load capability is reduced from 75 pounds to 33 pounds with increased run duration requirement from 1000 seconds to 2000 seconds since the 300 cycle operating time increases from 10 hours to 16.94 hours. The oxidizer bearing thrust load capability is reduced from 78 to 58 pounds with run duration increased from 1000 to 2000 seconds.

j. Throttling Requirement

The 25K thrust engine is basically developed for a throttle range of 5:1 but can without modification and the current design be throttled to 1/10 of the rated thrust level utilizing the 3 valve control system. The required valve settings are described in the off-design analysis. No change in start transient time, engine life, vehicle performance and envelope is anticipated.

Elimination of the throttling requirement can lead to a modification of the basic design. However, due to the relatively slow start transient and to permit thermal soaking of the turbine, throttle capability of a LO_2/LH_2 engine is always desirable.

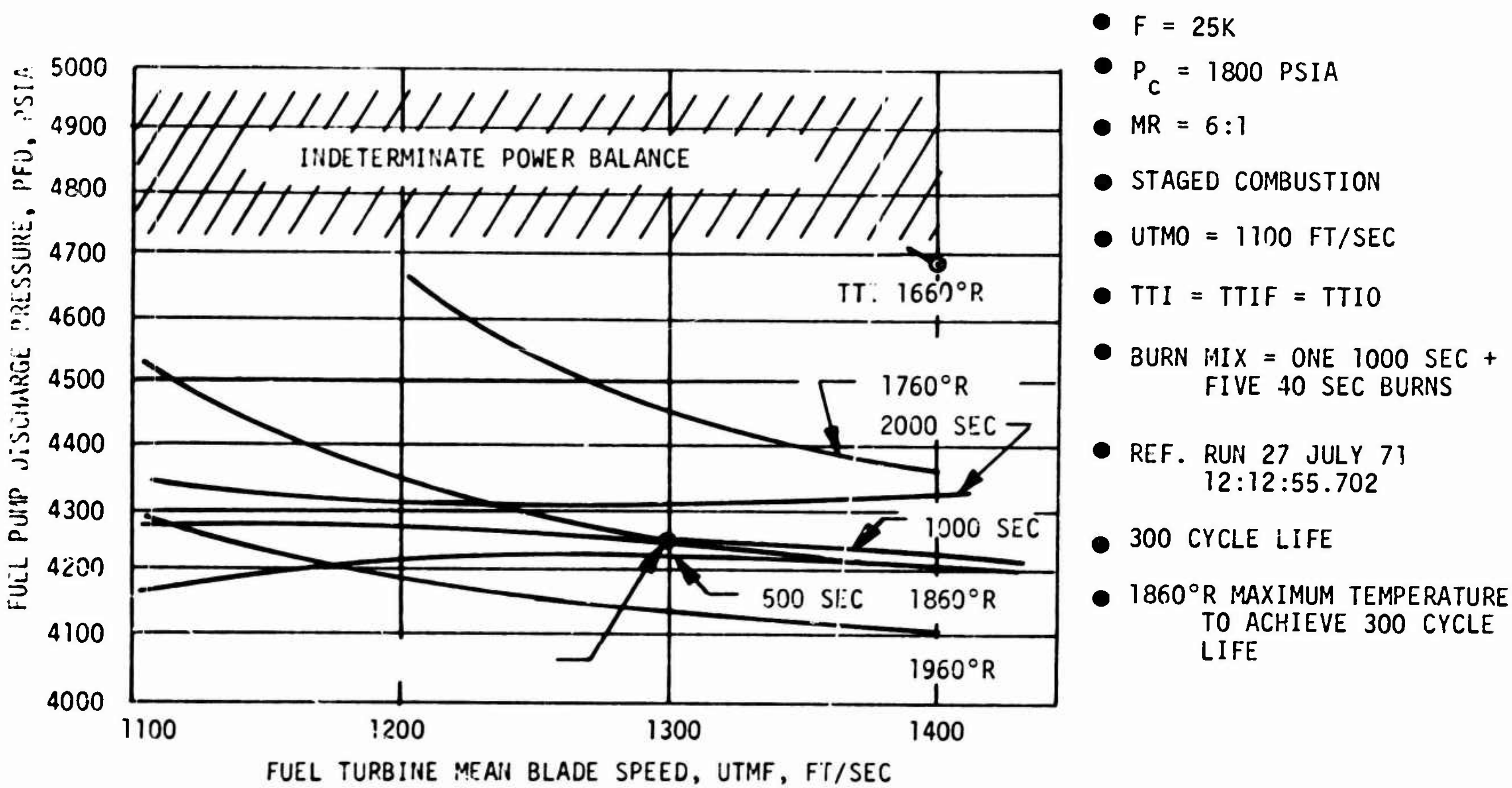
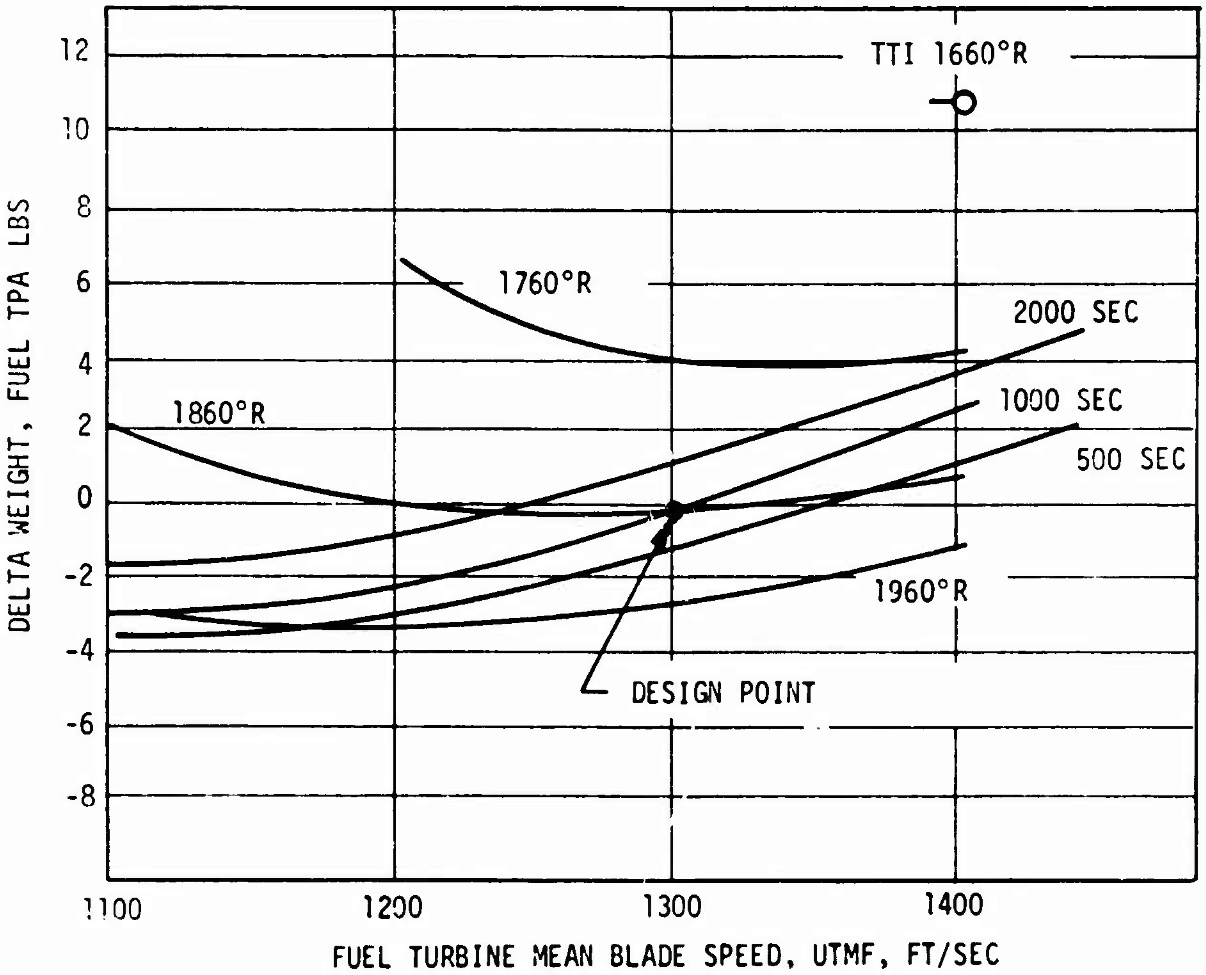


Figure 259. Single Burn Effect On Fuel Pump Allowable Discharge Pressure



P_c = 1800 PSIA

MR = 6:1

STAGED COMBUSTION

UTMO = 1100 FT/SEC

TTI = TTIF = TTIO

BURN MIX = ONE 1000 SEC +
FIVE 40 SEC BURNS

REF. RUN 27 JULY 71
12:12:55.702

300 CYCLE LIFE

1860°R MAXIMUM TEMPERATURE TO
ACHIEVE 300 CYCLE LIFE

Figure 260. Weight Sensitivity to Single Burn Duration - Fuel Turbopump

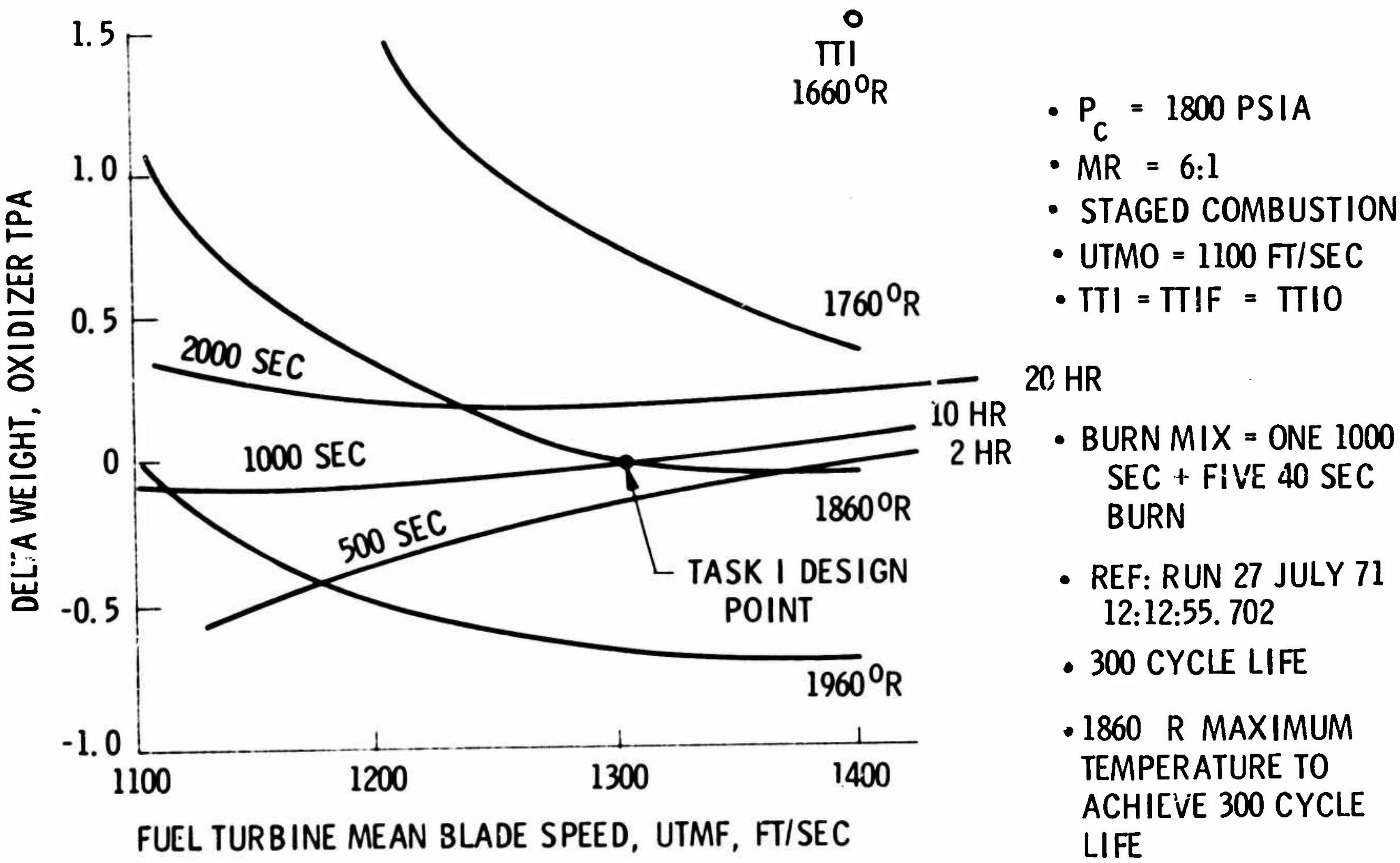


Figure 261. Weight Sensitivity To Single Burn Duration - Oxidizer Turbopump

TABLE LXXIV

TPA DURATION SENSITIVITY

OOS 25K ENGINE

TURBINE TEMPERATURE = 1860°R

	500 SEC		TASK I BASELINE 1000 SEC		REDUCE SHAFT AXIAL THRUST 2000 SEC		REDUCE SHAFT SPEED 2000 SEC	
	OXID	FUEL	OXID	FUEL	OXID	FUEL	OXID	FUEL
TURBINE INLET TEMPERATURE, °R	1860	1860	1860	1860	1860	1860	1860	1860
TURBINE MEAN BLADE SPEED, FT/SEC	1100	1380	1100	1300	1100	1230	1100	1230
PUMP DISCHARGE PRESSURE, PSIA	3092*	4220	3092*	4255	3092*	4315	3092*	4476
DELTA TPA WEIGHT, LB	-0.030Δ	+0.800Δ	REF.	REF.	+0.200Δ	-0.200Δ	+0.843Δ	+14.628Δ
DELTA BOOST PUMP WEIGHT, LB	0.0Δ	0.0Δ	REF.	REF.	0.0Δ	0.0Δ	0.0Δ	+0.279Δ
DELTA BOOST + TPA WEIGHT, LB	-0.030Δ	+0.800Δ	REF.	REF.	+0.200Δ	-0.200Δ	+0.843Δ	+14.907Δ
DELTA TPA LENGTH, INCHES	0.0Δ	0.210Δ	REF.	REF.	0.0Δ	-0.188Δ	0.0Δ	+1.220Δ
MAIN TPA SHAFT SPEED, RPM	50000.	80000.	50000.	80000.	50000.	80000.	50000.	70000.
THRUST BEARINGS MAXIMUM AXIAL FORCE (INCLUDES PRELOAD), LB	120+	120+	78	75	58	33	58	75
SUCTION LINE DIAMETER, INCH	3.135	3.604	3.135	3.604	3.135	3.604	3.135	3.769

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10 AUG 71 10:05:02.984

*FIRST STAGE PUMP

Due to chilldown propellant lead transient control requirements, modulating valves cannot be eliminated by single orifice control.

The engine modification considered for a fixed thrust engine configuration are:

. Elimination of the turbine and preburner bypass valve and line resulting in a 2 valve control system.

. Modification of the preburner injector and chamber for liquid oxidizer injector.

. Elimination of the LO_2 cooled section on the preburner and replacement by a fuel cooled section.

. Increase of the LO_2 preburner LO_2 circuit stiffness for liquid injection. to $\Delta P/P = 0.45$.

. Increase of the fuel pump discharge pressure to to accomodate increased injector stiffness. PFD = 4400 psia.

. Reduction of the fuel turbine tip speed to 1175 ft

. Incorporation of an autogenous LO_2 heat exchanger on the thrust chamber similar to the 10K engine design.

. Reduction of LO_2 preburner control valve pressure drop ΔP = 100 psia.

These modifications will not effect the normal engine performance, envelope, coolant requirements or engine life, but will result in a slight weight change and schedule change.

The increase of fuel pump discharge pressure requirement due to increase LO_2 preburner stiffness and preburner LO_2 valve modification are too small to have significant impact on engine weight. The following table indicates weight modifications:

Component	Weight Change, lb
Bypass Valve and Line	-13.1
Fuel TPA	0.
Ox. TPA	0.
Preburner Chamber	-2.0
Autogenous Heat Exchanger	3.0
Total Weight Change	-12.1 lb

The modified pressure schedule is shown in Figure 262.

Due to reduction of the preburner LO_2 valve pressure drop requirements, pump discharge pressures are modified only slightly such that the engine schedule is not significantly impacted.

k. Time in Orbit

Changing the engine storability requirements from 2 weeks up to 104 weeks has little effect on the baseline engine configuration specifically designed for 52 weeks of space storage time. Engine reusability requires that all valve seats are hard seats to avoid the cold flow problem of elastomeric seals.

No modifications are anticipated due to a change in space storage requirement.

l. Idle Mode

The base line engine is considered to be prechilled by propellant overboard dumping with pre-pressurized propellant tanks.

The engine idle mode is an operating mode where the engine operates under tank pressure only, without operation of the preburner and turbopumps. In this operating mode, the propellant tanks are considered settled and prepressurized to the nominal values.

The basic engine design can accept idle mode operation with the minimum of modification due to the three valve engine control concept. The turbine and preburner bypass valve which is full open at idle mode operation permits engine to operate without pump overspeed.

The modifications required are:

1. Incorporation of a modulating control valve in the thrust chamber oxidizer igniter circuit rather than on-off valve.

2. Main thrust chamber oxidizer valve has to be designed to give accurate metering in the thrust range of 0.5 to 3% of wide open admittance. This can be obtained by providing a small control valve in parallel to the main oxygen valve resulting in a small engine weight increase of GG. ΔW = 6 lb.

Figures 263 and 264 are schematic diagrams of the various idle mode operational sequences.

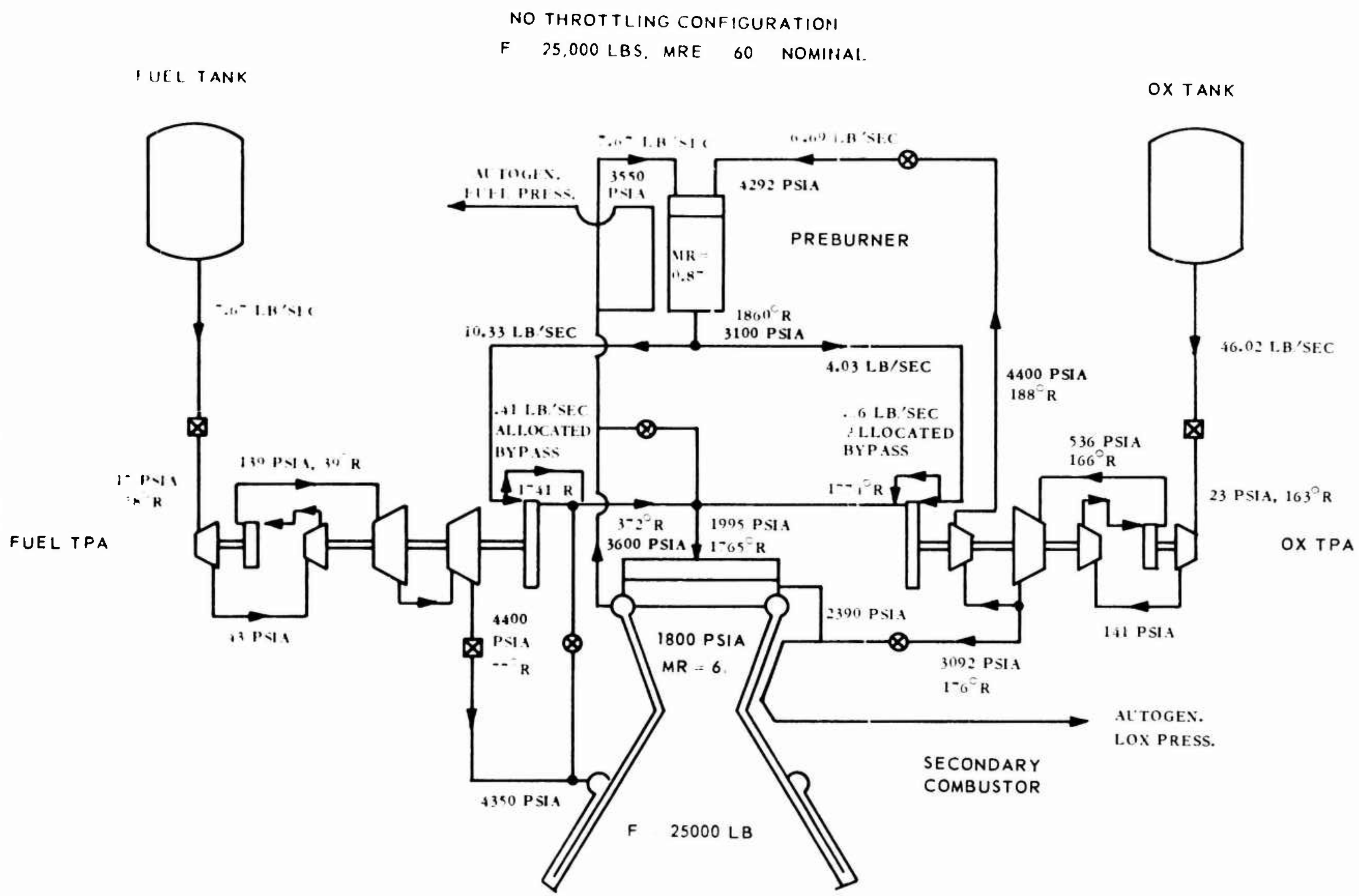
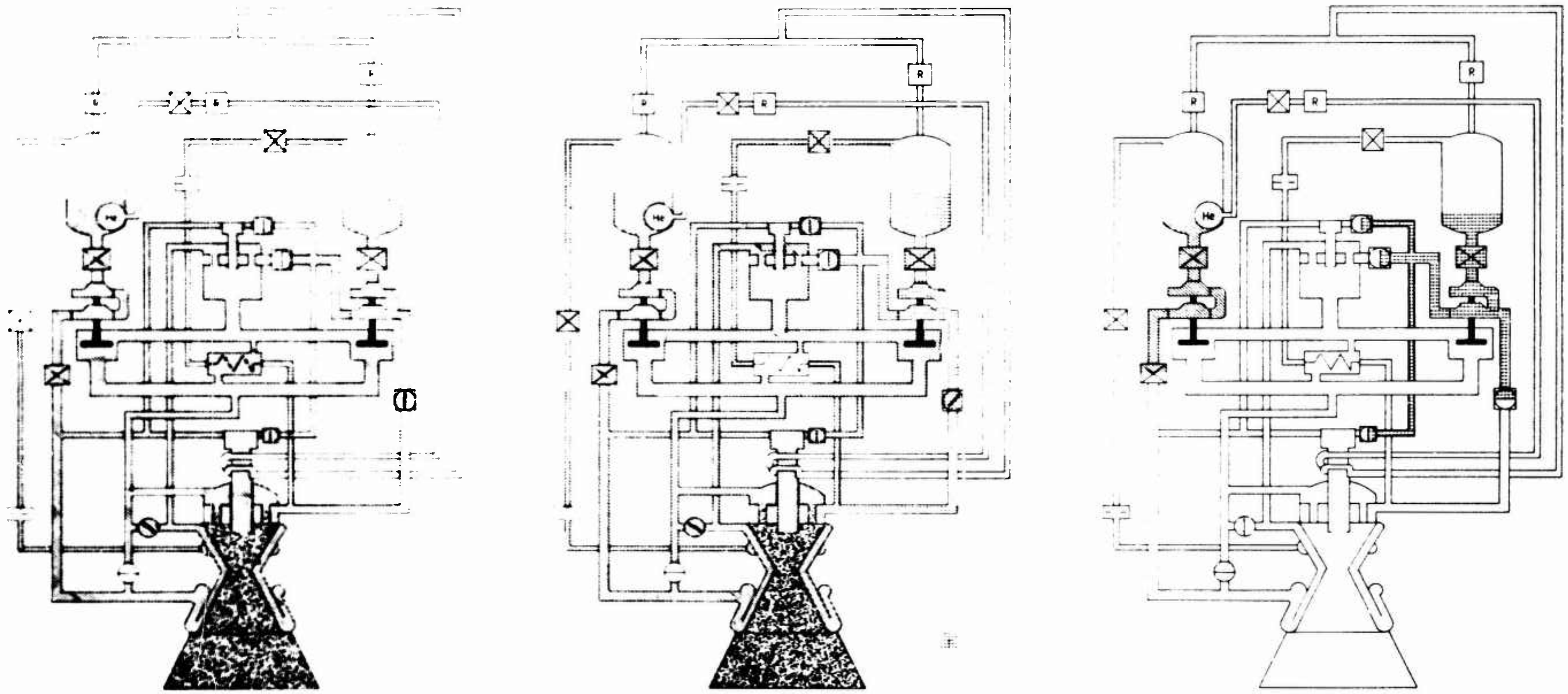


Figure 262. OOS Parameters for Engine Operation at Full Thrust - No Throttling Configuration



- PREBURNER OXID VALVE OFF
- He TANK PRESSURIZATION ON
- PREBURNER BYPASS VALVE OPEN
- PREBURNER IGNITER VALVE OFF

- OXID DISCHARGE VALVE MODULATE

- FUEL DISCHARGE VALVE CLOSED
- OXID DISCHARGE VALVE CLOSED
- He TANK PRESSURIZATION OFF

Figure 263. Post Engine Operation - Icle Mode

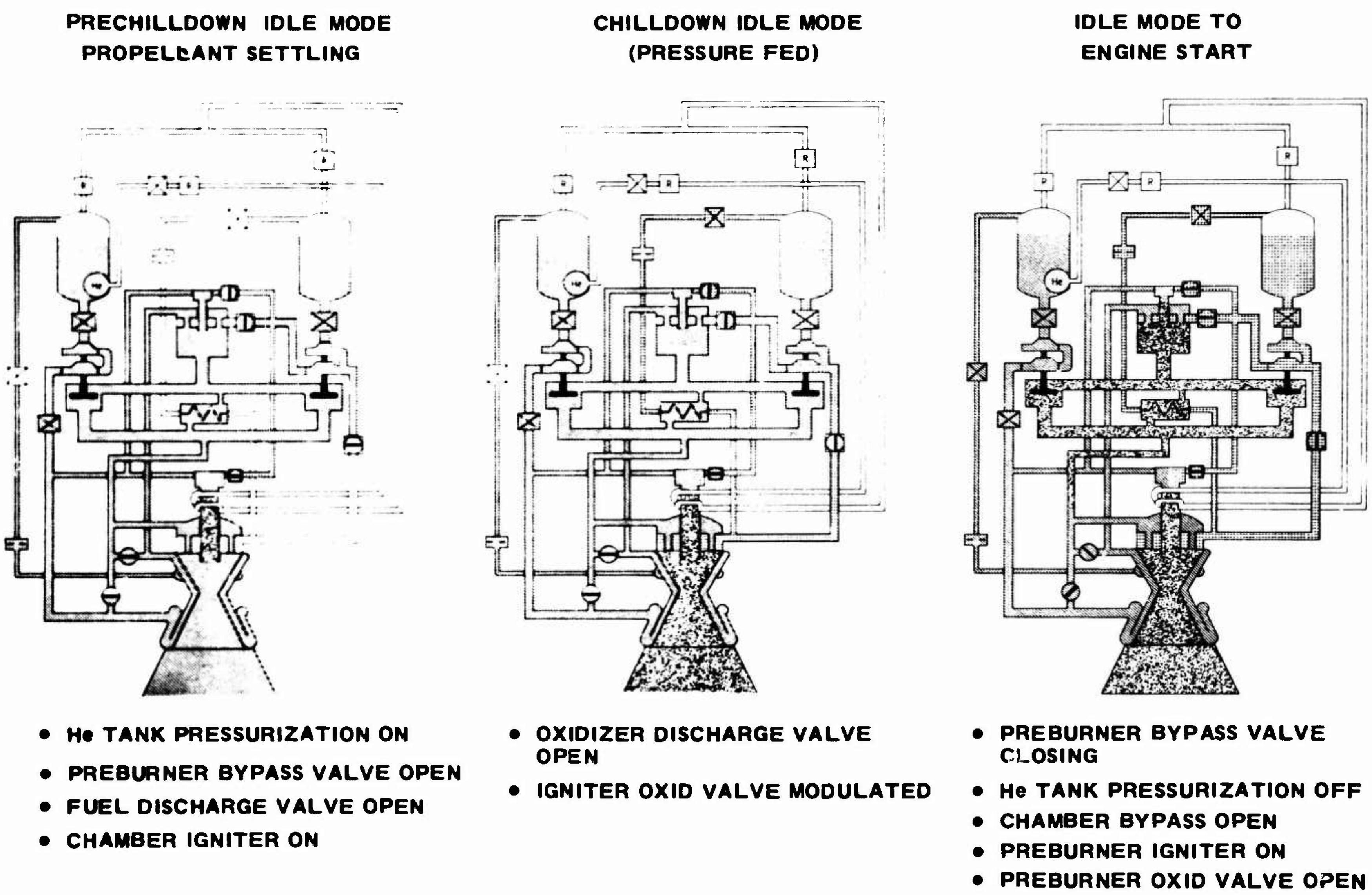


Figure 264. Idle Mode Operation

6. Engine Reliability Analysis

The OOS reliability effort was concerned with the following general areas:

1. Reliability program planning
2. Engine reliability growth
3. Relative cycle reliability
4. Failure modes and effects
5. System requirements for man-rated, reusable, unmanned and expendable engines

In order to utilize available contract funding in the most efficient manner, it was decided to emphasize the engine related studies because the reliability program plans are concerned primarily with implementing and coordinating procedures. This activity will be similar to the activity planned for other programs, and it is anticipated that these existing plans can be easily modified to satisfy the OOS requirements and objectives.

The work accomplished in each of these areas is as follows:

a. Program Plan

The Reliability Program Plan will describe the effort required to assure the achievement of reliability requirements for the ALRC OOS Engine.

This plan encompasses all reliability activities planned throughout the program. It is anticipated that the Reliability program plan will be maintained current by page revision or complete reissue, whichever is more cost-effective, to reflect all approved program changes.

Part A, the Program Description, will include discussions of the reliability management, engineering, and evaluation tasks to be performed during the program. The extent to which each requirement is to be implemented is described with reference to implementing procedures or documents.

Part B, Program Data, of the plan will include description of the data to be delivered to the Government in support of the reliability program. A description of the Reliability Data and Master File is also included to show the data to be used to support the OOS Program, and preparation of deliverable data.

All management, engineering, and operating personnel will exert an influence upon the reliability and, therefore, cost-effectiveness of the OOS engine. OOS management will assure that the requirements of the reliability program are clearly and specifically defined and that all reliability program tasks are planned and implemented in the most cost-effective

manner possible. These reliability program tasks will be evaluated and controlled in a manner consistent with their relative importance to the program and their demonstrated level of task effectiveness. The Reliability Plan provides for timely visibility into the status of the program and critical problem areas at all levels of management as well as at the pertinent engineering working levels. A system of reviews, audits, and surveys, and program plan corrections/revisions is maintained to assure the effectiveness of the reliability program throughout the OOS development and operational phases.

During development the reliability program tasks will be implemented using an integrated design team approach which will provide for cost-effective accomplishment of reliability tasks as well as the timely utilization of task results in the design and development process. Design specifications and drawings will be prepared by design teams utilizing the results of failure mode, effect, and criticality analyses performed at the system, subsystem, component, and part level for critical items. All design documentation will be reviewed by each engineering discipline. The design status documentation will provide management visibility by means of quarterly progress reports, joint ALRC/AFRPL management reviews, and critical design review mileposts. Design team analyses will include review of selected parts, materials, and design standards, including part application stresses and safety margins. Use of established design standards and previously qualified parts or components will be emphasized. Management visibility for this effort is provided through lists of preferred/nonpreferred parts and justification for electrical, electronic, and electromechanical items.

A closed-loop control system for problem/failure analysis and correction will be implemented beginning with the initial series of component or higher level prototype development tests. This will assure maximum growth in hardware maturity and a consequent reduction in total program costs. Problems/failures external to the engine design will receive the same degree of management visibility as design problems/failures. This will include test equipment, procedures, and personnel problems, hardware nonconformances, and human error related problems. Development and production problems/failures will require the same analysis and correction emphasis as will be required for problems/failures from field tests and/or flight tests depending upon their criticality classification and ranking. Monthly nonconformance status and trend summary reports will provide management with visibility into the effectiveness of the closed-loop and correction activities. Unfavorable critical problem/failure trends in the development and production programs will be highlighted by design team/component areas.

All design analyses and development test data will be used to establish the minimum maintainability requirements for each critical component to assure their continued reliable and safe operation over the full service life of the engine. These requisites will be identified early to provide the maximum amount of engine development data for analyses of FTC baseline maintenance requirements. Maintenance requirements will require cost effectiveness and technical justification. Maintenance Significant Items List and critical design review data will provide management visibility

into this very crucial area of the OOS program. Assessments of the maturity of the OOS design; quality control system; and test procedures, personnel, and equipment system will be provided monthly based upon analyses of the cumulative rate of premature test terminations due to these causes. Maturity data will be reported in a manner which clearly indicates the trends of each maturity index for management visibility. Recommendations for revisions to the design, development test program or maintenance requirements will be made if the trends are unfavorable for the objective of rapid maturity growth and low-cost, reliable operation.

The management authority and control of the reliability program will be maintained at ALRC, Sacramento, California. Reliability personnel will be located at other sites to monitor the implementation of this plan by affected site functions, assure visibility of site reliability problem areas, and provide site management with the direct reliability interface required to achieve reliability objectives in a timely and effective manner. All reliability data storage, engineering analysis, and deliverable data functions will be performed at ALRC, Sacramento. Nonconformance status and trend reporting, significant nonconformance reports, an lysis plans of action, maturity assessments, and quarterly status reviews and reports data will provide the required visibility to program management regarding the most critical operations and flight support reliability problem areas.

b. Reliability Growth

The reliability growth studies were conducted to determine the growth rate that can reasonably be expected during developmental and acceptance testing of the OOS engine. The results of the study were used in the establishment of a practical engine reliability goal, in the development of an engine test program, as an aid in defining peripheral engine testing requirements, and as a basis for monitoring reliability growth during development.

The results showed that an engine reliability of 0.99 can be expected to occur at about 777 tests. The use of peripheral tests can be expected to provide a saving of about 62 tests and to cause about 3 extraneous, peripherally-induced failures. Measures of the changes in these results are evaluated based on variations in certain reliability-affecting parameters. Component reliabilities are expected to range from 0.9987 to 0.9996 when the engine reliability is 0.99.

The methods of analysis used to obtain the foregoing information are summarized in the following paragraphs.

(1) Reliability Growth Model

The model for reliability growth is based on the following model that relates the expected cumulative number of failures to the cumulative number of tests:

$$F(t) = \frac{t}{A + Bt}$$

where

t = cumulative number of tests,

$F(t)$ = number of failures expected to be accumulated after completion of the $t^{\underline{th}}$ test, and A and B are positive constants that specify the reliability growth characteristics of the item under consideration.

The instantaneous unreliability (probability of failure) is the derivative of the above model:

$$U(t) = \frac{dF}{dt} = \frac{A}{(A + Bt)^2}$$

where

$U(t)$ = the unreliability of the item during the $t^{\underline{th}}$ test.

The instantaneous reliability (probability of success) is

$$R(t) = 1 - U(t)$$

where

$R(t)$ = the reliability of the item during the $t^{\underline{th}}$ test.

(2) Estimation of the Growth Parameters

The growth parameters, A and B, can readily be evaluated for any item (like the OOS engine) by means of a two-step analysis. First, parameter values are established for a similar item (like a Titan engine) for which sufficient historical data is available. This is usually accomplished by using the success rate method, which provides a maximum likelihood fit of the above model to the historic data. The second step is to perform an analysis to evaluate a numerical index of complexity (called the relative failure potential), which indicates the failure proneness of one item (the OOS engine) relative to another item (the Titan engine). Then the growth parameters for the OOS engine can be estimated as follows:

$$A = \frac{A'}{P}$$

$$B = \frac{B'}{P}$$

where

P .....relative failure potential of the OOS engine with respect to the Titan engine (i.e., ratio of expected unreliability of the OOS engine to that of the Titan engine at the same stage of development).

A, B....estimated growth parameters for the OOS engine.

A',B'...growth parameters established for the Titan engine based on applicable test data.

(3) Effects of Peripheral Testing

Peripheral testing (that is, testing at more severe conditions) can be used to accelerate the process of reliability growth, thereby reducing the number of tests required to attain a given reliability level. However, this saving is not obtained without penalty. More expensive test facilities may be required for peripheral tests than for nominal tests. Also, the direct cost of testing can be expected to be greater due to such considerations as increased propellant flow rates or extended test durations. Furthermore, some extraneous failures can be expected to occur because of new failure mechanisms that are introduced as a result of the more severe testing conditions.

When testing is changed from nominal to peripheral, the unreliability is increased in accordance with the following formula:

$$U_p = (1+\lambda)^a U_n$$

where

λ = the relative increase in the severity of peripheral tests over nominal tests,

U_n = the unreliability at nominal conditions,

U_p = the unreliability at peripheral conditions, and

a = an exponent.

For mechanical hardware, the value of the exponent ranges from about three to about nine, with the value $a = 6$ considered to be the most representative general value.

When testing is returned to nominal conditions from peripheral conditions, the inverse of the above relationship is used:

$$U_n = (1+\lambda)^{-a} U_p$$

The formulas for computing the number of tests saved by peripheral testing, the numbers of applicable nominal failures and extraneous failures incurred during the period of peripheral testing, and the actual reliability growth curve corrected to nominal conditions are complex and are considered beyond the scope of this report.

(4) The Titan Data Base

An analysis was performed on the test history for a single subassembly of the Titan II first stage engine. All design failures were included in the analysis. All other failures (manufacturing, quality control, human error, procedures, etc.) were excluded. The maximum likelihood fit of the data provided the following values of the growth parameters for the Titan engine.

$A' = 8.09342384$

$B' = 0.02471448$

(5) Relative Failure Potential

A relative failure potential analysis was performed for each of three candidate designs of the OOS engine. The relative failure potentials of those three systems with respect to a single subassembly of the first stage Titan II engine are as follows:

$P = 1.0273$ Staged Combustion Cycle

$P = 1.0273$ Staged Combustion Bleed Cycle

$P = 0.9337$ Expander Cycle

The value $P = 1.0273$ was used for most of the analyses because the staged combustion cycle is the primary candidate design.

(6) Parameter Values Used in the Analysis

Several values were used for each of the parameters introduced previously, as indicated as follows:

$\lambda = 0, 0.09, 0.12$ relative increase in test severity

$a = 3, 6$ exponent

$P = 1.0273, 0.9337$ relative failure potential

Calculations were performed for six different cases, as identified in the following table.

Case	P	λ	a
1	1.0273	0.09	6
1a	1.0273	0	-
2	1.0273	0.09	3
3	1.0273	0.12	6
4	0.9337	0.09	6
4a	0.9337	0	-

Cases 1a and 4a represent situations in which no peripheral testing is performed. In each of the other four cases, nominal testing is terminated on test 117 and resumed again on test 329. The intervening period of 211, from test 118 to test 328, inclusive, is devoted to peripheral testing.

(7) Results of Calculations

The results of the calculations are presented in Table LXXV for cases 1 to 3 and in Table LXXVI for cases 4 and 4a. The predicted reliability growth is shown in Figure 265 based on anticipated peripheral conditions, as represented by case 1. Case 1a is also shown (for reference purposes) by the dashed curve. Cases 1 to 3 are shown in Figure 266, providing a comparison that indicates the effect of changes in the parameter λ (relative increases in test severity) and a (the exponent). The effect that variations in P (relative failure potential) have on OOS engine reliability for λ = 0.09 and a = 6 is shown in Figure 264 corresponding to 800 tests. This plot covers cases 1 and 4. Table LXXVII indicates (for cases 1, 2, 3, and 4) the number of tests saved as a result of peripheral testing, the number of extraneous failures brought about by the more severe conditions of peripheral testing, and the number of nominal-level failures occurring during the period of peripheral testing.

(8) Component Reliability Apportionment

Based on case 1 and the relative failure potential analysis given in III,B,6,c, an OOS engine reliability of 0.99 (which is predicted to occur on engine test 777) is apportioned to the major components. The results are presented in Table LXXVIII. The component reliabilities were established such that their product is 0.99 and such that each component failure rate is proportional to the corresponding relative failure potential.

c. Estimate of Relative Reliability of OOS Engine Turbine Drive Cycles

(1) This analysis was made to obtain a relative reliability of three candidate engine cycles:

1. Expander cycle
2. Staged combustion bleed cycle
3. Staged combustion cycle

TABLE LXXV

PREDICTED OOS RELIABILITY GROWTH, FOR P = 1.0273

TEST NUMBER	CASE 1 $\lambda = 0.09$ $a = 6$ FAILURES	RELIABILITY	CASE 1a $\lambda = 0$ $a = 6$ FAILURES	RELIABILITY	CASE 2 $\lambda = 0.09$ $a = 3$ FAILURES	RELIABILITY	CASE 3 $\lambda = 0.12$ $a = 6$ FAILURES	RELIABILITY
10	1.231	0.8805	1.231	0.8805	1.231	0.8805	1.231	0.8805
100	9.723	0.9255	9.723	0.9255	9.723	0.9255	9.723	0.9255
117	10.941	0.9311	10.941	0.9311	10.941	0.9311	10.941	0.9311
118*	11.015	0.9314	11.010	0.9314	11.011	0.9314	11.017	0.9314
200	16.398	0.9535	15.760	0.9511	15.960	0.9518	16.597	0.9542
328**	22.600	0.9736	20.799	0.9683	21.347	0.9700	23.194	0.9752
329	22.627	0.9736	20.831	0.9684	21.377	0.9701	23.219	0.9753
400	24.330	0.9782	22.855	0.9743	23.300	0.9755	24.821	0.9794
600	27.831	0.9861	26.890	0.9842	27.165	0.9848	28.128	0.9867
800	30.125	0.9904	29.493	0.9893	29.680	0.9896	30.344	0.9907

* First peripheral test for all cases except case 1a.
** Last peripheral test for all cases except case 1a.

A = 7.87834502 .....Reliability growth parameters for all four cases
B = 0.02405770

TABLE LXXVI

PREDICTED OOS RELIABILITY GROWTH FOR P = 0.9337

TEST NUMBER	CASE 4, $\lambda = 0.09$, a = 6 FAILURES	RELIABILITY	CASE 4a, $\lambda = 0$, a = 6 FAILURES	RELIABILITY
10	1.119	0.8914	1.119	0.8914
100	8.837	0.9323	8.837	0.9323
117	9.944	0.9374	9.944	0.9374
118*	10.075	0.9380	10.007	0.9377
200	14.954	0.9579	14.324	0.9555
328**	20.541	0.9760	18.904	0.9712
329	20.565	0.9760	18.933	0.9713
400	22.113	0.9802	20.772	0.9766
600	22.279	0.9874	24.440	0.9856
800	27.380	0.9913	26.806	0.9903

* First peripheral test for case 4

** Last peripheral test for case 4

A = 8.66812021 .....Reliability growth parameters for both cases
B = 0.02646940

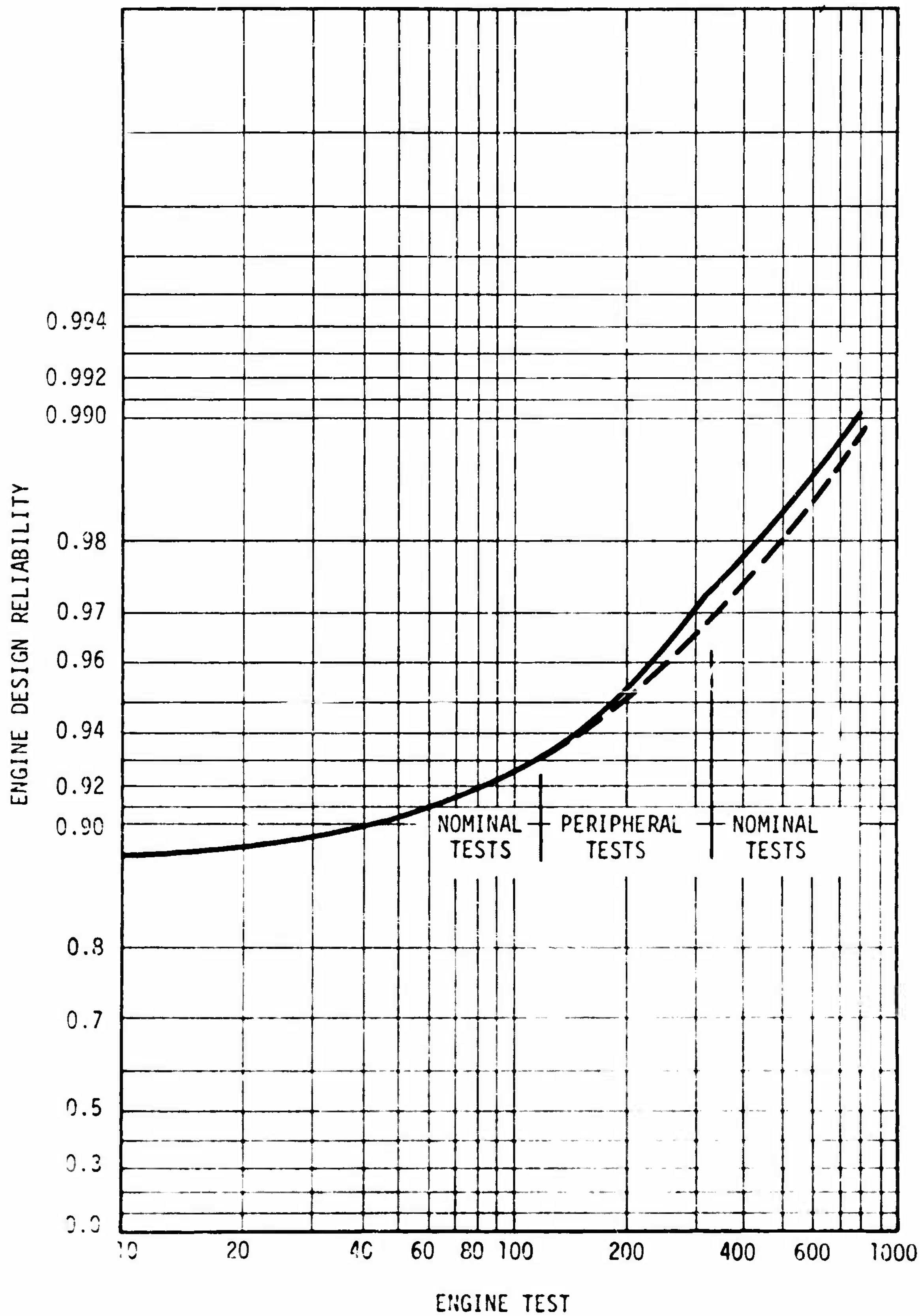


Figure 265. Predicted OOS Reliability Growth, Case 1

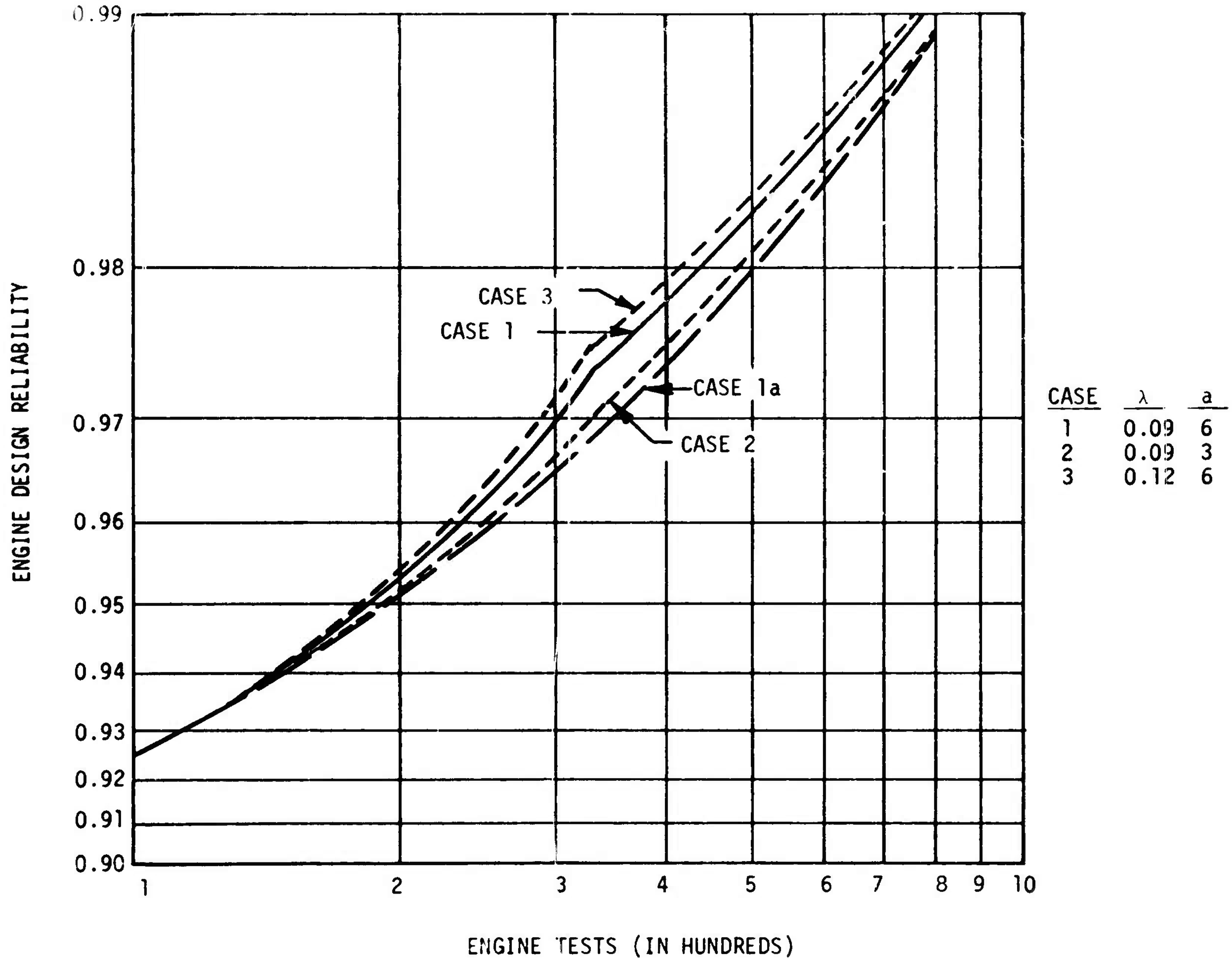


Figure 266. Comparative OOS Reliability Growth Characteristics

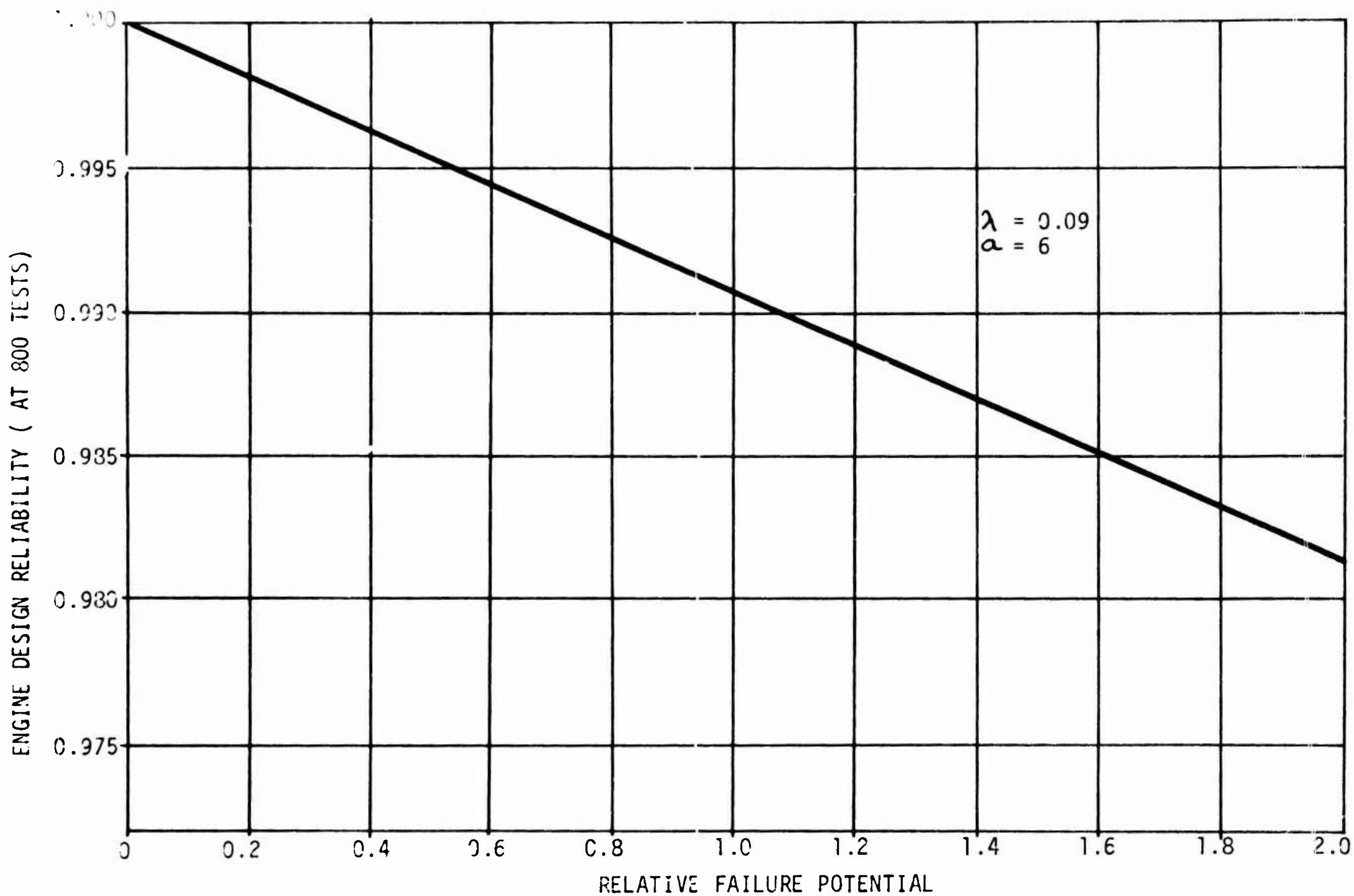


Figure 267. Effects of Variations in P on Reliability at 800 Tests

TABLE LXXVII

EFFECTS OF PERIPHERAL TESTING

Case	P	λ	a	Tests Saved	Extraneous Failures	Nominal Failures
1	1.0273	0.09	6	62	3.456	11.659
2	1.0273	0.09	3	18	1.442	10.406
3	1.0273	0.12	6	85	4.986	12.253
4	0.9337	0.09	6	62	3.100	10.534

TABLE LXXVIII

COMPONENT RELIABILITY APPORTIONMENT

Component Type	Number	Reliability
Oxidizer TPA	1	0.9990
Fuel TPA	1	0.9987
TCA and Igniter	1	0.9987
Valve	8	0.9996
Preburner and Igniter	1	0.9994
Heat Exchanger	2	0.9995
Lines	1	0.9991
Engine Controller	1	0.9993

The relative failure distributions of the OOS turbine drive cycles were estimated using Titan family engine failure data.

The Titan failure data used as a basis for the analysis is given in Table LXXIX.

The general procedure was to list those components which are common to the Titan and OOS systems along with the numbers of each component and the Titan relative failure rates (see the 1st, 2nd and 3rd columns of Table LXXX. Column 4 lists the numbers of components of the OOS expander cycle and column 5 lists the failure distribution based on the number of individual components relative to Titan. Column 6 is the failure distribution adjusted to a base of 100%. The remaining columns are repeats of columns 4, 5, and 6 but for the other two OOS cycles.

Since the Titan II and III engines do not have igniters, Titan I test data was used to obtain a ratio of gas generator igniter failures to total gas generator assembly failures. The ratio of thrust chamber igniter failures to total thrust chamber assembly failures was also obtained from the Titan I data. Table LXXXI summarizes this data.

Note on Table LXXX that there were no failures recorded for the gimbal assembly and TCA igniter.

A comparison of the columns headed "Failure distribution relative to Titan" shows that the OOS expander cycle is the most reliable of the three cycles. This is to be expected from this type of preliminary analysis since the expander cycle has one less pump stage and two less valves than the other two proposed cycles.

Note that the relative reliability between the staged combustion cycle and staged combustion bleed cycle is the same. This is due to the fact that there is no difference in the type and numbers of components between these two cycles.

d. Failure Modes and Effects

(1) Introduction and Summary

A preliminary failure modes and effects analysis was performed for the Orbit-to-Orbit Shuttle (OOS) Engine. The results of the analysis are presented in Table LXXXII, which lists the engine components and the engine subsystems in which they function, the required component functions, the anticipated component failure modes, the expected effects of those failures on the associated subsystem and engine, and numeric criticality rankings for the engine effects. The criticalities were based on the categories as listed in Table LXXXIII. The analysis was based on the staged combustion cycle engine.

TABLE LXXIX

COMPONENT RELATIVE FAILURE DISTRIBUTION

COMPONENT OR PART NAME	STAGE 1 SUBASSEMBLY	STAGE 2	TOTAL	AVERAGE	
TPA				27.92	
Oxidizer Pump	--	1.93	1.93		0.97
Fuel Pump	0.90	4.00	4.90		2.45
Gear Box	7.91	4.60	12.51		6.26
Turbine Assembly	13.80	20.00	33.80		16.91
Lube Oil System	--	2.66	2.66		1.33
Solid Start Cartridge	13.57	28.48	42.05	21.04	
TCA				13.75	
Combustion Chamber	12.66	11.27	23.93		11.98
Injector	2.94	0.60	3.54		1.77
Oxidizer Dome	--	--	--		--
Valves				12.85	
TCOV and Linkage	--	--	--		--
TCFV and Actuator	2.94	0.60	3.54		1.77
PSV and Solenoid	7.91	5.33	13.24		6.62
PSV O'brd Ck. Valve	0.90	--	0.90		0.45
GGD Ck V	4.75	3.27	8.02		4.01
GGF Ck V	--	--	--		--
Gas Generator Assembly	4.75	5.33	10.08	5.04	
Heat Exchangers				4.75	
Oxidizer Superheater	7.46	--	7.46		3.73
Gas Cooler Assembly	2.03	--	2.03		1.02
Lines				9.49	
Pump Suction (2)	--	--	--		--
Pump Discharge (2)	3.84	1.93	5.77		2.89
Vaned Elbows (2)	0.90	--	0.90		0.45
Oxidizer Bootstrap	6.72	3.27	10.05		5.03
Fuel Bootstrap	0.90	1.33	2.23		1.12
Engine Control System and Instrumentation	4.97	4.00	8.97	4.49	
Pressure Switch	--	1.33	1.33	0.67	
Gimbal Assembly	--	--	--	--	
Frame	--	--	--	--	
Roll Control Assembly				--	
Nozzle	N/A	--	--		--
Bearing	N/A	--	--		--
Total	99.91	99.93	199.84	100.00	

TABLE LXXX

PREDICTED RELATIVE FAILURE DISTRIBUTION FOR OOS ENGINE COMPONENTS

THREE PROPOSED ENGINE SYSTEMS

COMPONENT OR PART	TITAN ENGINES: NUMBER OF COMPONENTS	TITAN ENGINES: RELATIVE FAILURE DISTRIBUTION FROM TABLE LXXIV	EXPANDER CYCLE: NUMBER OF COMPONENTS	EXPANDER CYCLE: FAILURE DISTRIBUTION RELATIVE TO TITAN	EXPANDER CYCLE: FAILURE DISTRIBUTION RELATIVE TO OOS	STAGED COMBUSTION BLEED CYCLE: NUMBER OF COMPONENTS	STAGED COMBUSTION BLEED CYCLE: FAILURE DISTRIBUTION RELATIVE TO TITAN	STAGED COMBUSTION BLEED CYCLE: FAILURE DISTRIBUTION RELATIVE TO OOS	STAGED COMBUSTION CYCLE: NUMBER OF COMPONENTS	STAGED COMBUSTION CYCLE: FAILURE DISTRIBUTION RELATIVE TO TITAN	STAGED COMBUSTION CYCLE: FAILURE DISTRIBUTION RELATIVE TO OOS
OX TPA											
Oxidizer Pump	1 Stage	0.97	1 Stage	0.97	1.04	2 Stage	1.94	1.89	2 Stage	1.94	1.89
Turbine Assy	2 Rotors	16.91	1 Rotor	8.46	9.06	1 Rotor	8.46	8.23	1 Rotor	8.46	8.23
FUEL TPA											
Fuel Pump	1 Stage	2.45	2 Stage	4.90	5.25	2 Stage	4.90	4.77	2 Stage	4.90	4.77
Turbine Assy	2 Rotors	16.91	1 Rotor	8.46	9.06	1 Rotor	8.46	8.23	1 Rotor	8.46	8.23
TCA (Excluding Igniter)	1	13.75	1	13.75	14.73	1	13.75	13.38	1	13.75	13.38
Igniter	1	--	1	--	--	1	--	--	1	--	--
VALVES	2	8.29[1]	6	25.17	26.96	8	33.56	32.67	8	33.56	32.67
PREBURNERS	1	5.04	1	5.04	5.40	1	5.04	4.91	1	5.04	4.91
Igniter	1	0.89[4]	1	0.89	0.95	1	0.89	0.87		0.89	0.87
HEAT EXCHANGER	2	9.50	2	9.50	10.17	2	9.50	9.25	2	9.50	9.25
LINES	N[2]	9.49	N	9.49	10.16	N	9.49	9.24	N	9.49	9.24
ENGINE CONTROL SYSTEMS & INSTRUMENTATION	M[3]	4.49	1.5M	6.74	7.22	1.5M	6.74	6.56	1.5M	6.74	6.56
GIMBAL ASSY	1	--	1	--	--	1	--	--	1	--	--
TOTALS				93.37	100.00		102.73	100.00		102.73	100.00

NOTES:

1. This value is the sum of 1.77 (for the TCFV and Actuator) and 6.62 (for the PSV and Solenoid).
2. Measure of Line Numbers.
3. Measure of Control System and Instrumentation Complexity
4. This value is 3/17 of Preburner Value (5.04). The Ratio 3/17 is based on Titan I Test Data.

TABLE LXXXI

TITAN I GAS GENERATOR AND THRUST CHAMBER FAILURES

	TCA	GGA
Chamber Failures	14	8
Injector Failures	18	6
Igniter Failures	0	3
Total Failures	32	17
Ratio of Igniter Failures to Total Failures	$\frac{0}{32}$	$\frac{3}{17}$

TABLE LXXXII

FAILURE MODES AND EFFECTS ANALYSIS

... OOS ENGINE SUBSYSTEM PROPELLANT FEED

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
OXIDIZER Turbopump Assembly	Increase LO_2 pressure from tank discharge to provide: (1) Intermediate pressure LO_2 for: (a) Combustion with fuel-rich gas in main combustion chamber. (b) Pressurization of the LO_2 tank. (2) High Pressure LO_2 for: (a) Combustion with LH_2 in the preburner. (b) Combustion with LH_2 in the preburner igniter. (c) Combustion with LH_2 in the igniter for the main combustion chamber.	Reduced Pressure/Flow	Reduced power output from the preburner.	Reduced thrust and mixture ratio, which can be corrected by engine controller.	4	°Pump Suction pressure °Pump Discharge pressure °Oxidizer Flow rate °Pump Speed
		Low Pressure/flow	Low power output from the preburner	Low thrust and mixture ratio.	3	°Preburner Pressure
		Very low or complete loss of pressure/flow.	Very low or complete loss of power output from the preburner.	Failure to start. Premature shut-down due to loss of thrust and loss of pressurization of LO_2 tank.	Start: 2B Steady State: 2A	°Turbine Exit Pressure °Preburner Temperature
		Structural Failure	Fire Hazard	Fire Hazard.	1	Impeller-to-Housing clearance Purge flow to inter-bearing cavity.

TABLE LXXXII (cont.)

SYSTEM OOS ENGINE SUBSYSTEM PROPELLANT FEED

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
FULL TURBOPUMP ASSEMBLY	Increase LH_2 pressure from tank discharge to provide high pressure LH_2 for: (1) Combustion with LO_2 in the Preburner. (2) Cooling the turbine rotor and bearing in the oxidizer TPA. (3) Cooling of main combustion chamber and nozzle. (4) Pressurization of LH_2 tank. (5) Combustion with LO_2 in the preburner igniter. (6) Combustion with LO_2 in the igniter for the main combustion chamber.	Reduced Pressure/Flow.	Increased temp. of gas output from the preburner. Reduced hardware life.	Reduced thrust and increased mixture ratio, which can be corrected by the engine controller.	4	°Pump Suction Pressure °Pump Discharge Pressure °Fuel Flow Rate °Pump Speed °Preburner Pressure °Turbine Exit Pressure
		Low Pressure/ Flow.	High temperature of gas output from the preburner. Reduced hardware life.	Low thrust and high mixture ratio.	3	
		Very low or complete loss of pressure/flow.	Very high temperature of gas output from the preburner. Reduced hardware life.	Failure to start.	Start: 2B	
				Premature shutdown due to loss of thrust and loss of pressurization of LH_2 tank.	Steady State: 2A	

TABLE LXXXII (cont.)

ENGINE   SUBSYSTEM PROPELLANT FEED

DATE

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
PREBURNER AND INJECTOR	Supply fuel-rich hot gas to: (1) Turbine in the oxidizer TPA. (2) Turbine in the fuel TPA. (3) Oxidizer heat exchanger. (4) Fuel heat exchanger.	Erosion or streaking.	Reduced hardware life.	No performance anomalies. Need for more frequent maintenance.		
		Performance anomalies.	Possible reduction in hardware life.	Reduced engine thrust and shift in mixture ratio.	Start: 2B Steady State: 2A or 3	∘Preburner Pressure ∘Preburner Temperature

TABLE LXXXII (cont.)

OOS ENGINE SUBSYSTEM PROPELLANT FEED

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
PREBURNER IGNITER, Including Spark Plug Assembly	Provide hot gas required to initiate combustion in the preburner.	Fails to ignite	No ignition in the preburner.	Fails to start.	Start Only: 2B	∘Preburner Input Current ∘Preburner Pressure ∘Preburner Monitor Current

TABLE LXXXII (cont.)

OOS ENGINE   SUBSYSTEM PROPELLANT FEED   PAGE 5 OF 24
DATE

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
OXIDIZER THRUST VALVE CHAMBER	A two-position, normally closed valve that initiates and terminates LO_2 flow to the injector for the main combustion chamber and to the LO_2 heat exchanger	Fails to open.	No LO_2 flow in the main injector or the LO_2 heat exchanger.	Failure to start. No pressurization of LO_2 tank.	Start only: 2B	◦Valve Current ◦Valve Position ◦Oxidizer Pump Discharge Press. Thrust Chamber Side ◦Oxidizer Thrust Chamber Inlet Pressure
		Fails to close.	Failure to end LO_2 flow to main injector and LO_2 heat exchanger. Possible hardware damage.	Delayed shutdown with excessive thrust tail-off.	2A	
		Premature opening or closing: Not applicable because of characteristics of engine controller.	N/A	N/A	N/A	

TABLE LXXXII (cont.)

[illegible] ENGINE SUBSYSTEM PROPELLANT FEED PAGE 6 OF 24 DATE

[illegible] NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
FUEL PUMP DIS-CHARGE VALVE	A two-position, normally closed valve that initiates and terminates LH_2 flow to the main combustion chamber/nozzle coolant tubes, main CC igniter, preburner and igniter, LH_2 heat exchanger and the oxidizer TPA (turbine rotor and bearing coolant).	Fails to open.	No LH_2 flow.	Fails to start	Start only: 2B	◦Valve Current ◦Valve Position ◦Fuel Pump Discharge Pressure ◦Fuel Thrust Chamber Jacket Inlet Pressure
		Fails to close.	Failure to end LH_2 flow.	Delayed shutdown with excessive thrust tail-off.	2A	
		Premature opening or closing: Not applicable because of characteristics of engine controller.	N/A	N/A	N/A	

TABLE LXXXII (cont.)

SUBSYSTEM PROPELLANT FEED

ITEM	FUNCTION	FAILURE MODE	FAILURE EFFECT		MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
			SUBSYSTEM	ENGINE		
OXIDIZER PREBURNER VALVE	Controls Engine thrust by controlling LO_2 flow to the preburner.	Fails to actuate on command.	No control of LO_2 flow to the preburner.	No control of thrust.	1 to 4 (Depending on required thrust profile for the mission)	∘Valve Current ∘Valve Position ∘Oxidizer Pump discharge press preburner side ∘Oxidizer preburner inlet pressure.

TABLE LXXXII (cont.)

[illegible] OOS ENGINE SUBSYSTEM PROPELLANT FEED

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
COOLING JACKET BYPASS VALVE	A two-position, normally closed valve that by-passes LH_2 flow around the cooling tubes in the main combustion chamber and nozzle during the engine start transient period.	Fails to open.	Insufficient LH_2 flow to the preburner, resulting in insufficient hot gas flow to drive the two TPA's.	Fails to start.	Start only: 2B	∘Valve Current ∘Valve Position ∘Thrust Chamber Jacket Fuel Inlet Press. ∘Turbine Exit Pressure
		Fails to close.	None	Inadequate cooling of main combustion chamber and nozzle causing hardware damage.	Start only: 2B	
		Inadvertent opening during steady state operation: Not applicable because of characteristics of engine controller.	N/A	N/A	N/A	

TABLE LXXXII (cont.)

[illegible] OOS ENGINE   SUBSYSTEM PROPELLANT FEED   PAGE 9 OF 24
DATE

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
PREBURNER BYPASS VALVE	A two-position, normally closed valve that by-passes LH_2 flow from the exit of the cooling tubes in the main combustion chamber nozzle directly to the chamber injector during the engine start transient period.	Fails to open.	Cavitation of the fuel TPA.	Fails to start.	Start Only: 2B	∘Valve Current ∘Valve Position ∘Preburner Fuel Injector Inlet Press. ∘Turbine Inlet Pressure
		Fails to close.	None	Engine will not boot strap to full thrust.	Start Only: 2B	
		Inadvertent opening during steady state operation: Not applicable because of characteristics of engine controller.	N/A	N/A	N/A	

TABLE LXXXII (cont.)

...005 ENGINE		SUBSYSTEM PROPELLANT FEED				PAGE 10 OF 24 DATE
ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
PREBURNER IGNITER OXIDIZER VALVE	A two-position, normally closed valve that initiates and terminates flow to the preburner igniter.	Fails to open.	No LO_2 flow to the preburner igniter, causing failure of the preburner to ignite.	Fails to start.	Start Only: 2B	∘Valve Current ∘Valve Position ∘Oxidizer Pump Disch. Press.-Preburner side ∘Preburner Chamber Pressure
		Premature closure	Premature termination of LO_2 flow to the preburner igniter	Fails to start if closure immediately follows opening.	Start: 2B	
				None otherwise.	Steady State: 4	
		Fails to close.	LO_2 flow to the preburner igniter not terminated as scheduled, resulting in an increased mixture ratio in the igniter and possible hardware damage.	No performance anomalies. Need for more frequent maintenance.	4	

TABLE LXXXII (cont.)

[illegible] ENGINE SUBSYSTEM PROPELLANT FEED

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
MAIN CHAMBER IGNITER OXIDIZER VALVE	A two-position, normally closed valve that initiates and terminates flow to the igniter for the main combustion chamber.	Fails to open	No LO_2 flow to the main chamber igniter, causing failure of the main chamber to ignite.	Fails to start.	Start Only: 2B	∘Valve Current ∘Valve Position ∘Oxidizer Pump Disch. Press.-Preburner side ∘Thrust Chamber Pressure
		Premature closure.	None	Premature termination of LO_2 flow to the main chamber igniter. Engine fails to start if closure immediately follows opening. Otherwise, no effect.	Start: 2B Steady State: 4	
		Fails to close.	LO_2 flow to the main chamber igniter not terminated as scheduled, causing an increase in the mixture ratio in the main chamber igniter and possible hardware damage.	No performance anomalies. Need for more frequent maintenance.	4	

TABLE LXXXII (cont.)

SYSTEM OOS ENGINE SUBSYSTEM PROPELLANT FEED PAGE 12 OF 24 DATE

ITEM	FUNCTION	FAILURE MODE	FAILURE EFFECT		MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
DRAWING NO.			SUBSYSTEM	ENGINE		
HOT GAS MANIFOLD	Duct fuel-rich hot gas from the turbine exhausts of the fuel and oxidizer TPA's to the inputs to the main chamber injector and to fuel and oxidizer heat exchangers. Transmits loads from all gimbaled components to the gimbal.	No primary failure modes.	N/A	N/A	N/A	

TABLE LXXXII (cont.)

[illegible] ENGINE SUBSYSTEM PROPELLANT FEED

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DATE

[illegible] DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
LINES	Duct propellants from one component to another throughout the engine.	No primary failure modes	N/A	N/A	N/A	

TABLE LXXXII (cont.)

[illegible] OOS ENGINE SUBSYSTEM PROPELLANT FEED

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
OXIDIZER HEAT EXCHANGER	Heat a portion of the LO_2 output of the oxid. TPA to form GO_2 at the proper thermal condition for pressurization of the oxidizer tank.	Internal leakage at one or more of the multiple internal connections.	Fire/explosion hazard due to mixing of fuel-rich hot gas with LO_2 and/or GO_2.	Fire/explosion hazard	1	∘Oxidizer Pressurant Pressure. ∘Oxidizer Pressurant Temperature

TABLE LXXXII (cont.)

... ENGINE SUBSYSTEM PROPELLANT FEED

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DATE

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
FUEL HEAT EXCHANGER	Heat a portion of the LH_2 output from the fuel TPA to form GH_2 at the proper thermal condition for pressurization of the tank.	Internal leakage at one or more of the multiple internal connections.	None	None	4	∘ Fuel Pressurant Pressure ∘ Fuel Pressurant Temperature

TABLE LXXXII (cont.)

OOS ENGINE SUBSYSTEM THRUST CHAMBER

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
MAIN COMBUSTION CHAMBER AND NOZZLE	(1) Contains and directs combustion gases to provide thrust. (2) Provides regenerative tubes for: (a) Cooling chamber/nozzle walls by LH_2. (b) Heating the LH_2 before injection into the preburner. (3) Provides attachment points for the gimbal actuator clevises. (4) provides structural support for the nozzle extension and its deployment subsystem. (5) Transmits thrust loads developed in the nozzle extension.	Erosion or streaking.	Reduced hardware life.	No performance anomalies. Need for more frequent maintenance.	4	None
		Split tubes.	Reduced hardware life.	Minor performance anomalies.	4	∘Chamber Pressure ∘Fuel Thrust Chamber Jacket Inlet Press. ∘Fuel Preburner Inlet Press.

TABLE LXXXII (cont.)

[illegible] OOS ENGINE   SUBSYSTEM THRUST CHAMBER

ITEM			FAILURE EFFECT			
DRAWING NO.	FUNCTION	FAILURE MODE	SUBSYSTEM	ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
MAIN INJECTOR AND DISTRIBUTION PLATE	(1) Provide for uniform distribution of fuel-rich hot gas at injection plane.	Nonuniform distribution of hot gas or LO_2.	Erosion or streaking of injector face, main chamber and/or nozzle.	Minor performance anomalies. Need for more frequent maintenance.	4	o Chamber Pressure o Turbine Exhaust Pressure
	(2) Provide sufficient resistance to hot gas flow to cause the hot gas manifold to act as a plenum.		Split tubes in main chamber or nozzle.	Same as above.	4	
	(3) Inject LO_2 into the uniformly distributed hot gas flow in such a manner as to achieve a uniform mixture ratio at the injection plane.					

TABLE LXXXII (cont.)

SYSTEM OOS ENGINE SUBSYSTEM THRUST CHAMBER

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
MAIN CHAMBER IGNITER, Including Spark Plug Assembly	Provide Hot Gas required to initiate combustion in the main chamber.	Fails to ignite.	No ignition in the main combustion chamber.	Fails to start.	Start only: 2B	◦Valve Current ◦Valve Position ◦Oxidizer Disch. Press. - Prefurner Side ◦Turbine Exit Pressure ◦Chamber Pressure

TABLE LXXXII (cont.)

OOS ENGINE SUBSYSTEM THRUST CHAMBER

ITEM / DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
NOZZLE EXTENSION	Contains and directs combustion gases to provide more efficient use of those gases in producing thrust at altitude.	Nonuniform distribution of film cooling gas (obtained from the fuel and oxidizer heat exchangers).	Reduced hardware life.	No performance anomalies. Need for more frequent maintenance.	4	None

TABLE LXXXII (cont.)

SUBSYSTEM THRUST CHAMBER

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DATE

[illegible]	FUNCTION	FAILURE MODE	FAILURE EFFECT		MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
			SUBSYSTEM	ENGINE		
[illegible] EXTENSION [illegible] ASSEMBLY	(1) Provides structural support for the nozzle extension.	Fails to extend.	Nozzle extension not fully extended. Hardware damage.	Reduced thrust.	2A	
	(2) Provides the means to extend and retract the nozzle extension.	Fails to Retract.		Possible loss of vehicle/crew.	1	

TABLE LXXXII (cont.)

[illegible] ENGINE ___ SUBSYSTEM STRUCTURAL SUPPORT

[illegible] DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
THRUST STRUCTURE	Transmits the thrust developed by the engine in the vehicle, and provides support for various engine components.	None	N/A	N/A	N/A	

TABLE LXXXII (cont.)

ENGINE SUBSYSTEM STRUCTURAL SUPPORT

[illegible] NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT		MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
			SUBSYSTEM	ENGINE		
GIMBAL	Transfers loads from all gimbaled engine components to the vehicle, and provides dual axis gimbaling of the engine in response to forces from the gimbal actuators	None	N/A	N/A	N/A	

TABLE LXXXII (cont.)

ENGINE SUBSYSTEM PURGE PAGE 23 OF 24 DATE

ITEM NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT: SUBSYSTEM	FAILURE EFFECT: ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
PURGE SUBSYSTEM	(1) Separates the LH_2 from the LO_2 in the oxidizer TPA.	Loss of purge to the oxidizer TPA.	None	Possible fire/ explosion in the oxidizer TPA, causing damage to other engine/vehicle hardware.	1	
	(2) Removes residual propellants and contaminants from all engine components.	Fails to purge or terminates prematurely (pre-fire)	None	None, but launch aborted.	Pre-fire: 2B	
		Fails to terminate during engine start.	High pressure propellants back-flow into purge subsystem and vent overboard.	Low performance, with excessive propellant consumption.	2A or 3	
		Fails to purge or terminates prematurely. (post-fire)	None	Possible damage to components, decreasing life capability and requiring more frequent maintenance.	Post-fire: 4	

TABLE LXXXII (cont.)

... OOS ENGINE SUBSYSTEM ENGINE CONTROL

ITEM DRAWING NO.	FUNCTION	FAILURE MODE	FAILURE EFFECT SUBSYSTEM	FAILURE EFFECT ENGINE	MISSION PHASE AND CRITICALITY	FAILURE DETECTION METHOD
CONTROLLER	Monitors critical parameters and initiates engine shutdown in the event of failure Provides electrical interface with the vehicle. Controls engine thrust and mixture ratio. Controls start and shutdown sequences.	Loss of electrical power.	Fails to function	Fails to start or shuts down prematurely. (Fail-safe mode) Fails to start or continues to fire if already started. (Fail-operational mode) Loss of control of thrust and mixture ratio. Shutdown occurs upon propellant exhaustion.	Start: 2B Steady State: 2A Start: 2B Steady State: 1 or 2A	
	Performs on-board checkout.	Loss of input and/or output signals. Controller designed to preclude such anomalies.	N/A	N/A	N/A	

TABLE LXXXIII

FMEA CRITICALITY CLASSIFICATIONS

Category	Potential Effect of Failure
1	Loss of life of crew member(s).
2A	Immediate mission flight termination. Loss of primary mission objectives.
2B	Mission (or launch) cancellation.
3	Mission (or launch) delay, or loss of secondary mission objectives.
4	None of the above.

(2) Discussion

For the purposes of this analysis, the OOS engine was partitioned into the following five subsystems:

1. Propellant feed
2. Thrust chamber
3. Structural support
4. Purge
5. Engine control

In Table LXXXII each subsystem is analyzed in terms of its constituent components. The components are listed in the first column of the tabulation, and their functional requirements are identified in the second column. Potential component modes of failure are listed in the next column, followed by an identification of the effects of such failures on the functioning of the subsystem and engine, columns four and five. The last column is used for numeric criticality rankings in accordance with the five categories specified in Table LXXXIII.

Two primary failure modes were generally excluded from consideration in the analysis: structural failures and leakage. Structural failures were excluded for two reasons:

1. It is planned to use adequate margins of safety during the design process to preclude the occurrence of any such failures.

2. Component, subsystem, and engine tests will be used to verify the structural adequacy of the hardware.

Thus, structural failures are not expected to occur. On the other hand, leakage can reasonably be expected to occur in every component. The effect of such leakage on the engine and the criticality of that effect would depend on the severity of the leakage. Thus, leakage was excluded simply to eliminate the repetitious entries that would otherwise be found in Table LXXXII. However, the exclusion of leakage and structural failures from the tabulation should not be interpreted to mean that they are not being considered, for those failure modes will be very carefully considered and evaluated during design, development, and production.

There are two exceptions to the general exclusions mentioned above. One is a structural failure in the oxidizer turbopump assembly due to an explosion caused either by metal-to-metal rub in the oxidizer environment or by failure of the inert purge fluid to maintain the required separation of the fuel and oxidizer. The other exception is internal leakage in the oxidizer heat exchanger which would result in mixing of fuel and oxidizer and could cause an explosion and extensive hardware damage. These two [illegible] nature as to warrant special attention and emphasis.

Three basic operational characteristics were considered as a part of the analysis: fail-safe and fail-operational capabilities, and man-rating. The fail-safe capability is achieved in two ways:

1. The on-off valves are designed to fail closed, thereby resulting in a safe engine shutdown in the event that the supply of electric power to the engine controller is lost.

2. The controller is designed to shut down the engine in the event of a component malfunction.

The fail-operational capability can also be achieved with the same basic design by designing the controller to shift to a less severe operating point (if possible and safe to do so) in the event of a component malfunction. From Table LXXXII, the only components that can reasonably be expected to fail during steady state and require a significant adjustment in the engine operating point (item 2 above) are the turbopumps, preburner, two control valves, and oxidizer heat exchanger. Failures of these components may not permit continued safe operation of the engine. It may be possible to improve the fail-operational capability of the engine with respect to failures of those components by providing component redundancy. However, any gain achieved with such an approach is offset by several disadvantages:

1. More complex plumbing between components.
2. A need for more valves.
3. A more complex engine controller to monitor and control engine and component operation.
4. Additional development cost and time to demonstrate the adequacy of the redundancy scheme and its control.
5. More complex maintenance requirements.
6. Greater engine weight.
7. More complex malfunction diagnosis.

It would probably be better to avoid the use of redundancy (except in the engine controller) in order to attain the goal of a lightweight design. Instead, the individual components should be designed somewhat more conservatively in order to reduce the probability of failure to an acceptable level, with a minimal increase in engine weight. In a like manner, man-rating can be attained with the basic design by means of a more conservative design, rather than the use of redundancy. In addition, parts used for engine build-up must be carefully tested and screened to assure the delivery of engines of the highest reliability (similar to the approach used for Gemini), and a comprehensive program of scheduled and unscheduled maintenance must be established to maintain the man-rated status of the engine.

7. Configuration Variation Due to Projected Use

The engine system requirements were considered for man rated, reusable non-manned, and expendable engines. The recommended systems for each of these engines differs primarily in the control systems and instrumentation used. The control requirements for each of these systems is presented in Table LXXXIV through Table LXXXVI and summarized in Table LXXXVII.

It should be noted that the work statement required that the engine design be capable of being man-rated. Consequently, no major design configuration changes will be required to transform the engine from non-man rated to man-rated. Additional testing could be required for the purpose of demonstrating reliability. The scope and type of additional reliability testing will depend upon the demonstrated reliability at that time considered in conjunction with a mission analysis. As discussed in the previous section, the engine is now assumed to have a man-rated capability upon the successful completion of approximately 800 engine level tests.

TABLE LXXXIV

CONTROL SYSTEM RECOMMENDATIONS - MANNED REUSABLE MISSION

Parameter	Control System: Vehicle Mounted	Control System: Engine Mounted	Reason
Mixture Ratio	Adaptive Control Level Sensor	Flow Rate Measurement Flowmeters - Primary	Mission Flexibility
			Pilot Safety by Showing Progressive Failure
		Flow Rate Measurement by Boost Pumps - Backup	Problem Isolation
Thrust	Ground Set Level Command	P_c Feedback Control	Mission Flexibility
	Override Capability		Self Calibrating
	$\frac{\Delta V}{\Delta t}$ Control		
Sequencing	On-Off Command	Sequencing and Sequencing Control (Engine Mounted Controller)	Simultaneous Control and Engine Development Program
	ΔV Cutoff		

TABLE LXXXV

CONTROL SYSTEM RECOMMENDATIONS - UNMANNED MISSION

Parameter	Control System: Vehicle Mounted	Control System: Engine Mounted	Reason
Mixture Ratio	Progressive Control Ground Set	Flow Meter Feedback Control	Engine Performance Repeatability for Progressive Control Problem Isolation Self Calibrating System
Thrust	Ground Set	P_c Feedback Control	Exact Mission Timing Repeatability for Progressive Control
Sequencing System	On-Off Command ΔV Cutoff	Sequencing and Sequence Control (Engine Mounted Controller)	Engine Development Program

TABLE LXXXVI

CONTROL SYSTEM RECOMMENDATIONS - EXPENDABLE MISSION

Parameter	Control System: Vehicle Mounted	Control System: Engine Mounted	Reason
Mixture Ratio	Progressive MR Control Ground Set	Precalibrated Valve Positions	Simpler System Reduced Instrumentation No Flow Measurement Required Lower Development Cost and Time Small Propellant Utilization Variations Not Critical to Mission
Thrust	ΔV Cutoff Ground Set	Precalibrated Valve Settings	Small Thrust Variations Not Critical to Mission Reduced Instrumentation Lower Development Cost and Time
Sequencing System	On-Board Computer Multiplex System	None	Less Expensive System

TABLE LXXXVII

CONTROL SYSTEM SUMMARY

Engine Type	Engine Controller	Vehicle Controller	Engine Operation Data into Storage	Instrumentation Requirement	Main Control Parameters	Remarks
Man Rated	Yes	Yes	Yes	Redundant per Failure Modes Analysis Requirements (See Section IIIB-7)	1. Propellant flow rates and thrust 2. Primary engine and component operating parameters.	1. Potential malfunction can be detected and compensated for. 2. Vehicle controller will be used to back up Engine Controller for pilot safety. 3. Timely determination of component degradation for maintenance and mission safety. 4. Stored engine data used to determine mission maintenance requirements.
Unmanned Reusable	Depending on Maintenance Requirements	Yes	Yes	Non redundant and reduced to operating parameters only. No immediate malfunction detection.	1. Δ Velocity 2. Propellant consumption 3. Primary engine operating parameters.	1. Engine operating requirement determined by vehicle ΔV. 2. Prior to starts, level sensors are checked and data used to predetermined valve position for mixture ratio. Mixture ratio not measured during firing. 3. Stored engine data used to determine post mission maintenance requirements.
Expendable	No	Yes	No	No redundancy and major engine and component parameter only.	1. Δ Velocity 2. Propellant consumption.	1. Engine operation requirement determined by vehicle ΔV. 2. Prior to starts, level sensors are checked and data used to predetermine valve positions required for mixture ratio not measured during engine operation.

8. Improved Technology Impact

In the course of the OOS Engine Design Study, it became evident that more information was required than was available in certain areas of technology. In some cases insufficient detail data prevented accurate analyses. In other cases, it was discovered that significant increases in vehicle payload capability could be obtained by use of designs requiring advances in the current state-of-the-art. In these instances, assessments were made of the type and magnitude of the benefits to be derived as well as the nature of the technological advancement required. The most rewarding, and therefore, recommended, technology areas which need work are listed and discussed below:

1. Regenerative cooled thrust chamber LCF life vs environment in a high pressure O_2/H_2 engine.
2. JANNAF performance predictions for O_2/H_2.
3. Performance of very large area ratio nozzles.
4. Radiation cooled nozzle extension.

Reusable small engines require a considerable amount of instrumentation to permit efficient maintenance. Current instrumentation techniques and sensors contribute greatly to increase weight. Sensor miniturization would greatly help to reduce engine weight.

Engine maintenance requires additional flight readiness, and data recording, failure or performance degradation analysis instrumentation. An engine mounted computer forming an integral engine component should be developed in parallel with the engine.

Another maintenance requirement is that the engine be hot fire checked and at sea level duplicating actual engine operation. This is very problematic for large area ratio nozzles and may require new nozzle technology, in particular, radiation cooled light weight nozzle extensions which can be separated at low area ratio for sea level testing.

The operational requirements for multi-restart engines demands chilldown through idle mode. High pressure engines are somewhat handicapped because idle mode thrust levels are rather low due to the small throat size of these engines. These engines would greatly benefit from a pump assisted idle mode resulting in doubling idle mode thrust levels. Pump assisted idle modes may present mixture ratio control problems due to 2 phase flow and should be developed in an experimental program.

Engine dynamic stability was considered by the inclusion of injector baffles and acoustic dampers in both the main and preburner injectors. Both of these concepts are relatively recent technology and need extension.

The requirement of autogenous tank pressurization during pump assisted idle mode was a new operating requirement and requires demonstration. Particularly, its benefit for the LO_2 tank are questionable due to

the heavy weight of the oxidizer pressurant as compared to bottled helium which may result in large propellant outage.

Engine start transients were kept slow to minimize the thermal stress on the turbine disks and blade, and technology could be developed to lower these transient stresses by special disk designs permiting faster starts and improved impulse capability for start impulse operations.

The greatest need for more technology information was found to exist in the determination of thrust chamber material low cycle fatigue properties. The chamber fatigue life is dependent upon the total strain imposed upon it by the thermal gradients established during operation. The inner surface of the chamber is heated by convection from chamber gases; the backside is cooled by cryogenic hydrogen. The resulting thermal gradient (and material strain) can be minimized most effectively by holding the hot gas side wall temperature to a low value. This improves the low cycle fatigue properties of the material, and more importantly, minimizes the wall temperature differential. Maintenance of a low internal wall temperature requires very high coolant velocities in the regenerative coolant passages because of the extreme heat fluxes present near the throat of the thrust chamber. High coolant velocities result in large coolant pressures losses, and high coolant inlet pressures. Considering that the gas side heat flux increases with chamber pressure, so that the coolant pressure drop increases, the fuel coolant pump power requirement increases sharply with chamber pressure. Therefore, even with a staged combustion engine cycle, which provides a large amount of TPA power, engine power balance considerations limit the usable chamber pressure. The required pump discharge pressure also affects the weight and complexity of the turbopumps.

High chamber pressure engines have several desirable features, which include high thrust weight ratio, and high performance/length and diameter ratios. For a given chamber life, therefore, it is necessary to know the TCA material low cycle fatigue properties accurately in order to accomplish a realistic design study. Where the engine study includes low thrust, long life engines (as in the subject study) these effects are pronounced. The high cooled surface area/flow rate ratio of small thrust engines (especially with large area ratio, actively cooled nozzles) increase the heat load per pound of coolant, and increase the pressure losses.

Low cycle fatigue data of copper based thrust chamber alloys gathered to date at various agencies and contractors, including ALRC, have not shown good agreement with each other. It has recently become apparent that, as discussed in Appendix B, the environmental gaseous composition can have an order of magnitude, or greater, effect on low cycle fatigue life, all other conditions being equal.

It seems, for example, that the influence of strain hold time on fatigue life may be a much greater function of surface oxidation than internal creep damage. Selection of optimum thrust chamber material, then, as well as the entire engine design and specifications, are dependent upon

knowledge of material properties under the actual conditions of operation. It is recommended, therefore, that a materials evaluation program be conducted in which the atmosphere be accurately controlled, as well as the temperature, strain range, time, etc., as is normally done.

Accurate analytical determination of OOS class engine specific impulse performance is not possible with current technology. There are several reasons for this: Firstly, the JANNAF performance prediction method has not been standardized. There are at least two alternate methods, both of which have been used on the subject engine; they predict performances differing by as much as 2 seconds or greater. Under these conditions it is not profitable to expend much analytical effort on performance, because any result is in doubt.

It is obvious that the use of large area ratio nozzles improves system performance for OOS class vehicles. Optimization is difficult, however, because of the dearth of analytical/test data at area ratios of 100 or greater. Some difficulty was experienced in generating optimum Rao nozzle contours for area ratios in excess of 250, because there had not heretofore been a need for such data. In addition, there is no available test data to calibrate the calculated nozzle performance. Since the nozzle weight and size increases rapidly with area ratio, and the performance decrement falls with increased area ratios, the overall vehicle performance is greatly affected by the nozzle design. Consequently future studies must be considered theoretical until high nozzle area ratio performance technology is obtained.

The practicality of a large nozzle engine is in doubt unless such factors as fabrication, development, handling, and maintenance are considered along with nozzle weight, performance, and envelope. For example, it is possible that 0.010 in. wall thickness coolant tubes can be fabricated for the 25K engine nozzle, and they would effect an improvement in engine weight of 35 pounds. However, experience in handling experimental Titan nozzles, has shown that a very high damage rate exists with such items. It is believed that a reusable, maintainable engine cannot afford the luxury of excessively frangible components.

In the interests of weight, cost, and reliability, the subject baseline nozzle has no intermediate joints between the exit plane and the attachment to the thrust chamber. The flight engine, therefore, must be tested in an altitude facility, since the overall nozzle area ratio of 290 provides extremely low exit pressures, and is not designed for atmospheric operation. If an altitude facility is not available, or is too costly to use, a shorter nozzle must be fitted for ground testing. This alters the engine fuel system parameters, because significant fuel heat addition occurs in the large total wall area of the nozzle coolant passages. The engine might be provided with a warmed mixture of hydrogen, liquid and gas, effecting the preburner mixture ratio. The nozzle/thrust chamber joint of the baseline engine is of furnace brazed design to minimize weight and leakage. Posttest

fabrication sequences to attach the flight nozzle would probably invalidate the engine test data. The baseline engine design is testable, but it may not be found practical.

The use of a radiation cooled nozzle extension has several benefits. Firstly, it can be detached for sea level testing without upsetting the coolant system or operating parameters. If attached at a low area ratio, sea level testing can be accomplished with confidence, with available steam ejection facilities, or even to sea level back pressure. Secondly, significant engine weight savings can be effected, if a low attachment area ratio is used. Thirdly, the availability of cooler coolant to the thrust chamber, allows long life designs with less coolant pressure loss, and lighter engine designs. Finally, single wall, non-coolant carrying skirts are more easily handled, and are less sensitive to slight damage than regenerative units.

Studies conducted under the subject contract show that fibrous graphite nozzles are the weight-optimum, radiation cooled skirt candidate. The useful attachment area ratio, and joint-induced performance loss is a function of the joint design. It is, therefore, recommended that technology be established in the areas of:

(1) Fibrous graphite low cycle fatigue properties.
(2) Fibrous graphite coatings.
(3) Fibrous graphite joint design.
(4) Supersonic gaseous H_2 injection into exhaust nozzles.

Figure 268 illustrates a typical non-film cooled fibrous graphite/tube bundle nozzle joint design.

The pump fed low thrust engine appears to have some difficulties with miniturization and TPA's suffer from large internal leakage losses effecting the power balance and the efficiency potential of multi-stage high speed pumps. The rolling contact bearings are very small and operate at high DN values with the result that they must operate at very light loads relying on good thrust balancing. Bearing retainers have to be of low profile to permit adequate cooling and very limited experience is available on the cage life of small cryogenic bearings.

The bearings not only have to meet the life requirement but also the multi restart and thermal cycle requirement and very little data is available in the bearing sizes considered for the OOS engine thrust range.

As an alternate approach for TPA bearing technology it is recommended to consider hydrostatic bearings.

A hydrostatic bearing has the capability of achieving operational life in excess of 20 hours by virtue of the fact that there is no surface-to-surface contact during steady state operation. A fluid film-hydrostatic bearing will experience surface-to-serface rubbing during start-up

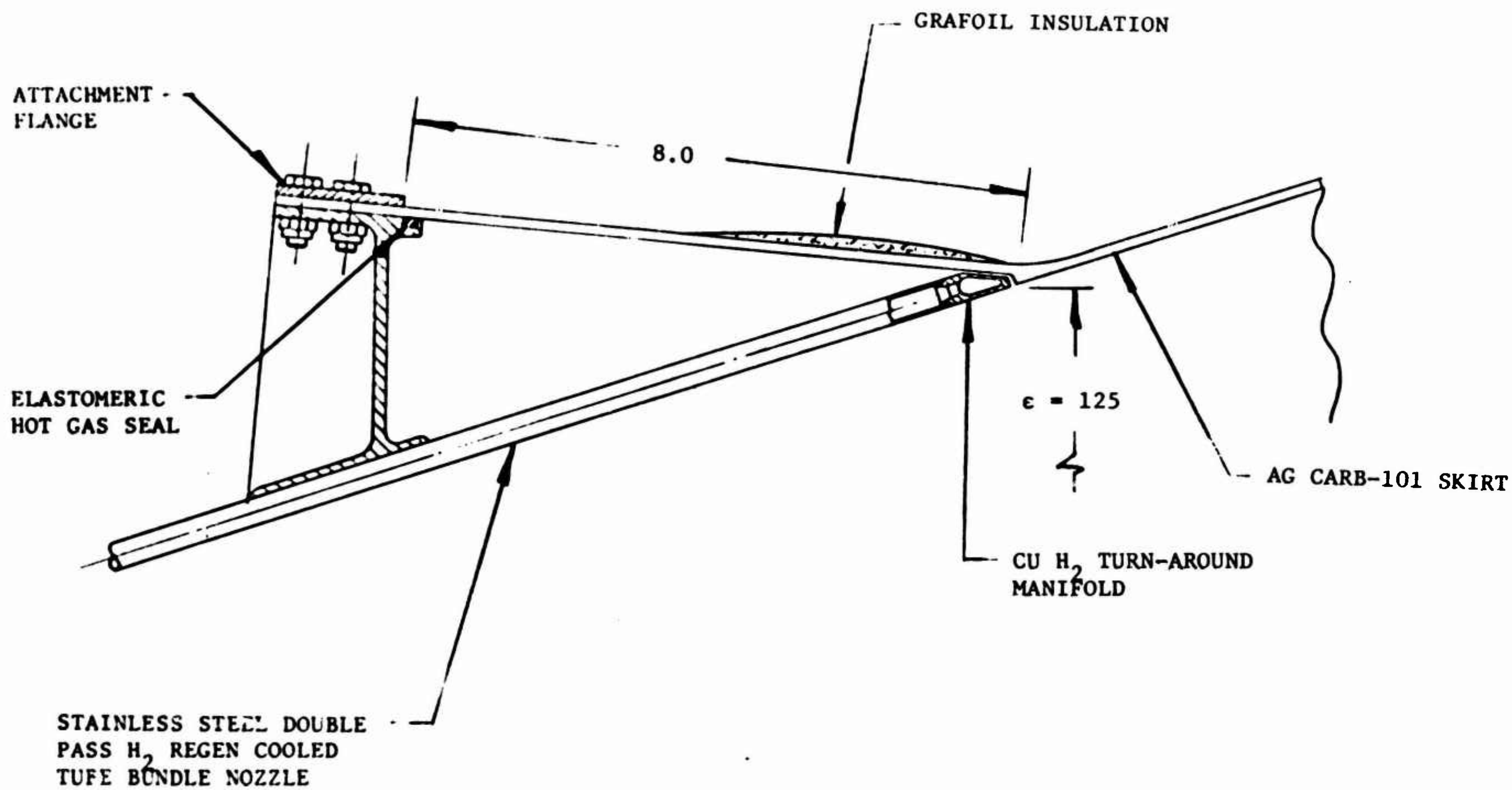

Figure 268. Radiation/Conduction Cooled Joint - Radiation Cooled Skirt

until system pressure levels reach a value to provide lift-off in the hydrostatic bearing. Limited data currently exists from bearing and seal testing to suggest that hydrostatic bearings would deliver the required 300 to 600 slow start cycles. No data is available to evaluate the feasibility of hydrostatic bearings in cryogenic propellant to accommodate fast start operation.

Existing LO_2 turbopumps with fuel rich hot gas driving the turbine, incorporate interpropellant seals utilizing a purge fluid for separation of the potentially reactive fluids.

An advanced triple vent interpropellant seal concept is proposed to eliminate the purge requirement from the interpropellant seal. This concept takes advantage of the low pressure of space to produce a sufficiently low pressure in the middle of the three vent cavities to preclude the possibility of ignition and combustion of the GO_2 - GH_2 mix.

There are no areas within the controls concepts, considered for the OOS Engine, that may be classified as new technology. There are, however, certain areas of technology which in the future could effect the component designs. These areas would include; 1) An elastomer material that has compliance and resilience at cryogenic temperatures for use as shutoff seals, dynamic shaft seals, and static seals. Development of this material would probably lead to replacement or modification of the current approach. 2) A readily castable material having good low temperature characteristics for high pressure applications. An ideal material for this application is Inconel, however, it cannot be reliably made as a casting. 3) Lighter weight materials having high magnetic properties for use in electrical actuator motors. This would result in smaller and lighter motors for a given power output.